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A STUDY OF STRATIFIED CHARGE FOR LIGHT DUTY POWER PLANTS

Volume 1

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16. ABSTRACT

The objectives of this project were to determine the acceptability of various types of stratified charge engines as potential power plants for light duty vehicles and motorcycles in America. The light duty vehicle considered was a 4/5 seat compact sedan with good acceleration capabilities and exhaust emissions below a primary target of 0.41 g/mile HC, 3.4 g/mile CO, 1.5 g/mile NO_x. A secondary target of 0.41 g/mile HC, 3.4 g/mile CO and 0.4 g/mile NO_x was also considered.

A literature survey was undertaken, comparing stratified charge engines with examples of good conventional gasoline and diesel engines. While some stratified charge engines had exhaust emission or fuel economy advantages, there were always sacrifices in other areas.

Eleven engines were configured, four of which were specifically directed towards the secondary emission targets. A method of rating the engines was derived, and the design concepts were compared with two gasoline engines by a jury panel. The overall result was that the Ford PROCO and Honda CVCC combustion processes were serious contenders to the gasoline engine at the primary emission target, and that both of these systems, together with the VW combustion process, might be suitable at the secondary targets.

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by

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Ann Arbor, Michigan 48105

October 1974

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GENERAL SUMMARY

This report describes a study which was conducted on behalf of the Office of Mobile Source Air Pollution Control of the Environmental Protection Agency and which was aimed at assessing the potential benefits and effects of using stratified charge engines as light duty power plants in the United States.

The theoretical advantages of stratified charge engines have been known for many years, particularly the potential for good fuel economy, low CO and NO_x emissions and some multi-fuel capability. The prospect of achieving the advantages has encouraged many organisations to develop their own stratified charge concepts. However, in spite of these advances only one stratified charge engine (Honda CVCC) has appeared in large numbers in motor vehicles. One aim of this project was to identify the reasons for the limited exploitation of the stratified charge concept, to determine the problem areas which were inhibiting further development, and to identify the stratified charge combustion systems which might benefit from further study, and could meet the short and long term requirements for automotive vehicles.

The first stage of the project involved a study of published literature from technical journals and learned societies throughout the world. The various stratified charge engines uncovered were then classified into seven general categories. Each category was studied separately, and many were found which could meet the proposed emission targets for vehicles in primary forms, i.e. 0.41 g/mile HC, 3.4 g/mile CO and 1.5 g/mile NO_x and at the secondary target 0.4 g/mile NO_x. However, with each system there were penalties compared to the gasoline engine in some of the following areas:

Specific torque and power output, noise, engine first cost, engine weight and bulk, odour and particulates. In addition, the improvement in fuel economy of the various stratified charge engines compared to good gasoline engines of today, was fairly small, usually less than 10%.

The next stage of the study involved the preliminary design of potentially viable stratified charge power plants. These power plants had to have adequate power to handle a 1600 kg (3500 lb) car with an acceleration equal to that of a "standard" American car essentially 96 kW (128 bhp). The concepts studied included reciprocating and rotary stratified charge engines, both with and without turbocharging. Two gasoline power plants were also outlined for comparison purposes. The configuration study covered the following contenders for the power plants.

1) 8 cylinder American gasoline engine meeting primary emission targets.

2) 6 cylinder European gasoline engine meeting primary emission targets.

3) 8 cylinder N/A engine, PROCO combustion system meeting primary emission targets.

4) 6 cylinder N/A engine, PROCO combustion system meeting primary emission targets.

5) 8 cylinder N/A engine, PROCO combustion system meeting secondary emission targets.

6) 8 cylinder N/A engine, TCCS combustion system meeting the primary emission targets.

7) 6 cylinder T/C engine, TCCS combustion system meeting the primary emission target.

8) 2-bank rotary engine, Curtiss-Wright combustion system meeting the primary

emission targets.

9) 8 cylinder T/C engine, TCCS combustion system meeting the secondary emission targets.

10) 8 cylinder N/A engine, MAN-FM combustion system meeting the primary emission targets.

11) 8 cylinder N/A engine, VW combustion system meeting both the primary and secondary emission targets.

12) 8 cylinder N/A engine, Honda CVCC combustion system meeting the primary emission targets.

13) 8 cylinder N/A engine, Honda CVCC combustion system meeting the secondary emission targets.

Drawings were prepared for each of these engines and the performance and exhaust emissions of each were estimated.

The third stage of the study was an assessment of the relative merits and viability of power plants which met both the primary and secondary targets.

In order to compare the various powerplants, a quantitative rating method was devised. This involved an estimate of the relative importance of 26 performance aspects and assigning to them numerical "weightings". Assessment of the merit with which each power plant met an aspect allowed a numerical rating to be assigned. Summation of the weighted rating allowed the power plants to be compared numerically. Use of this rating system on the power plants indicated that the gasoline engines as well as the Ford PROCO and Honda CVCC systems were all attractive at the primary emission target, and that the Ford PROCO, Honda CVCC and VW systems were viable at the secondary targets. However, the overall performance of all the engines at the secondary target was inferior to those at the primary target.

The fourth and final stage of the study was an assessment of the implication of the above results so that proposals for further work and research could be made.

It was concluded that:

1) HC emissions are a fundamental problem of unthrottled stratified charge engines which may not be completely soluble, but could be partially alleviated by further development work. Since unthrottled engines have the best fuel economy, alternative forms of initiating combustion other than by spark ignition should be studied.

2) Work should be concentrated on the Honda CVCC and Ford PROCO systems for the primary emission target.

3) Work should be concentrated on the Ford PROCO, Porsche and VW systems for the secondary emission target.

4) The MAN-FM combustion system should be applied to an automotive vehicle, to assess its potential, since emission and fuel economy test bed results are encouraging.

5) Research work is needed to derive expressions for heat transfer and fuel droplet motion and evaporation so that better mathematical models can be derived.

GENERAL INTRODUCTION

Concern with environmental issues such as exhaust emissions and energy utilisation of internal combustion gasoline engines has recently prompted an interest in alternative powerplants. The most attractive power plant would undoubtedly be one which could improve these aspects and involve the least change from the conventional engine. Consequently, variants of the spark ignition engine have received considerable study, and in particular those arrangements classed under the very general heading "stratified charge".

In the 1920s, Sir Harry Ricardo first postulated that an engine which was not throttled, and where the power was controlled by the fuel flow alone, would give the best possible fuel consumption from an internal combustion engine. The fuel would be distributed within the combustion chamber such that an ignitable mixture was formed at the spark plug under all engine operating conditions. At first, this definition of stratified charge was accepted, but over the years the term has been generalised to cover all spark ignition engines, either throttled or unthrottled, where the fuel is not distributed homogeneously throughout the combustion chamber.

It can easily be shown from theoretical considerations, that the compression ratio of an unthrottled stratified charge engine can be chosen to give the best possible fuel consumption. It can also be shown that such an engine will have lower CO and NO_x emissions than throttled gasoline engines. These considerations have prompted many experimenters to make new explorations of the field of stratified charge, and many new engines have emerged within the past few years. However, due to certain fundamental problems with the stratified charge concept, only one example (the Honda CVCC) has been applied to an automotive application, in large scale production.

This survey was commissioned in an attempt to draw conclusions from the unco-ordinated developments of many research groups, each working on different types of stratified charge engines, and to make recommendations as to which areas should receive more intensive effort. The envisaged applications of these engines was to be in passenger cars and motorcycles. Furthermore, the compromises between various aspects of the engine performance, such as between emissions and fuel economy, were to be identified. Finally, those combustion systems which might form viable automotive powerplants were to be schemed in an engine design, and rated in order of preference, by a technical review panel.

SUMMARY OF LITERATURE SURVEY

This portion of the report contains a digest of the important information which has been extracted from the literature survey. The first section contains an explanation of the derivation of baseline performance, emission and fuel economy data for conventional gasoline and diesel engines, against which the alternative stratified charge engines were to be compared. The conventional engines chosen as the baseline were considered to represent good engines of the present day. All stratified charge engines have been divided into 7 general categories, such that all the engines in one class have a similar combustion principle, and thus similar characteristics. This approach has simplified the treatment within the text and allowed specific conclusions to be drawn under each category.

The overall fuel economy advantage of the best stratified charge engines proved to be relatively small, compared to good examples of current gasoline engines. This was partly caused by trade-offs between exhaust emissions and fuel economy, and partly by fundamental limitations in the whole concept of stratified charge. Some types of engines showed no advantage in fuel economy over the baseline gasoline engine, although in some, exhaust emissions were easier to control.

Stratified charge engines are generally inferior to the gasoline engine in, at least some, of the following aspects; specific power output, noise, particulate and odour emissions, engine weight and volume and production costs. However, all the engines reviewed had advantages in CO and NO emissions. Many engines, when fitted in vehicles, met the primary target of 0.41 g/mile HC, 3.4 g/mile CO, 1.5 g/mile NO_x by application of exhaust treatment with thermal reactors and catalysts, and some could meet the secondary target of 0.4 g/mile NO_x. However, in achieving the secondary targets, there were always severe sacrifices in at least two of the following three performance aspects; specific power output, driveability and fuel economy.

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INTRODUCTION TO LITERATURE SURVEY

The purpose of the literature survey was to review stratified charge engine developments extending from 1920 through to the present day, on a world-wide basis. The main source for input information was published literature from the technical press, and patents. The aim of the survey was to provide information on all aspects of the performance of each combustion system, so that engine designs could be schemed where applicable, and the overall characteristics of the final target vehicle could be estimated.

Throughout this report the analysis and presentation of the published data has been orientated towards the target vehicle, and a summary of the specification of this vehicle is included here.

The vehicle and application considered was a 1600 kg (3500 lb) 4/5 seat sedan capable of achieving 0 to 97 km per hour (0 to 60 miles per hour) in 15.5 seconds, 40 to 113 km/hr (25 to 70 mph) in 15 seconds and the D.O.T. high speed pass manoevre in 15 seconds so that an installed power of about 96 kW (128 BHP) was required.

The emission targets were that the vehicle should achieve the following figures measured by the CVS/CH test procedure.

(a) Primary targets

HC	0.41 g/mile
CO	3.4 g/mile
NO$_x$	1.5 g/mile

(b) Secondary targets

As above but

NO$_x$	0.4 g/mile

A secondary objective of this study was to consider the applicability of stratified charge combustion systems to motorcycle applications in general. Specific configurations were not considered although for simplicity, the literature survey often considers the extreme ranges of motorcycle engines, i.e. the economy bike with a 50 to 200 cc cylinder size and the super bikes of 750 cc. The emission targets for this application were:

HC	a sliding scale between:	
	50 - 170 cc	8 g/mile
	170 - 750 cc	8.0 - 22.4 g/mile
	> 750 cc	22.4 g/mile
CO	fixed level	27 g/mile
NO	fixed level	1.92 g/mile

Scope of the Survey

In considering the various types of stratified charge engines, the results of the literature survey have been included under twelve general topic areas as described below. In many instances through the report, critical information on a particular topic area of an engine was missing from the literature, and under these conditions the probable characteristic has been postulated from the detailed knowledge of other aspects of the engine.

Topic Area (a) Combustion Chamber and Stratification Technique

This section includes a description of the concept and layout of the combustion system and the method by which the stratification of the fuel is achieved. Under a number of sub-headings, the specification of the various engine components needed to achieve that stratification are described. These sub-headings usually include the intake swirl requirements, the fuelling equipment (i.e. fuel injection or carburettor design), the ignition equipment and spark plugs, and the amount of intake throttling used. Finally, the text specifies any limitations on the compression ratio of the combustion system.

Topic Area (b) Combustion Characteristics & Heat Rejection

In some cases, studies of the combustion in stratified charge engines have been undertaken by combustion photography and the recording of cylinder pressure diagrams. This type of study often indicates the ways in which the combustion has diverged from that conceived by its inventor and sometimes reveals the source of problem areas. Heat rejection to the coolant is also considered in this section.

Topic Area (c) Performance

This section considered the maximum specific torque output (i.e. BMEP) recorded by stratified charge engines and the principal factors affecting the power output, i.e. limitations on engine speed range imposed by the combustion system or the engine components, volumetric efficiency, engine friction levels, and turbocharging. Where available, vehicle performance results and driveability are also considered.

Topic Area (d) Fuel Consumption

In this section the various fuel consumption results recorded from test beds and vehicles are discussed.

Topic Area (e) Exhaust Emissions

The study not only included test bed and vehicle HC, NO_x and CO emission results, but also information on other emissions, such as smoke, particulates, odour, aldehydes and ammonia where available. Emission control techniques such as EGR and the application of catalysts are also important, as is the durability of these components. Attempting to meet the secondary emission targets often influenced vehicle driveability and this aspect is considered here.

Topic Area (f) Multifuel Capability & Manufacturability

The multifuel capability of the stratified charge concept was not considered to be of immediate importance in choosing viable systems for the configuration study of an engine for the target passenger car. However, this aspect is obviously important in other applications, and may even become important in the context of passenger cars at some time in the future.

Topic Area (g) Engine Components

The stratified charge concepts usually impose additional engine complexity, such as fuel injection equipment and control arrangements for air fuel ratio. The reliability, durability and maintenance requirements of these items are discussed.

Topic Area (h) Cost

In certain cases, estimates have been made for changing passenger car engines from the standard gasoline type to a stratified charge concept. This information is included where available, but in many cases there was no data, and this particular topic area has been omitted. To cover this omission, Ricardo have made their own production cost estimate of each of the configured stratified engines in volume 2 of this report.

Topic Area (i) Mathematical Models

In view of the interest and lack of understanding of stratified charge engine combustion and NO_x formation, this area received particular emphasis.

Topic Area (j) Engine Size and Weight

Some existing gasoline engines have been converted to various stratified charge combustion systems, and the resulting effects on engine size and weight are tabulated in this section. The conversion may affect engine height by requirements for piston cavities, or weight by the inclusion of a turbocharger, or extra ancillaries such as vacuum pumps for vehicle braking.

Topic Area (k) Noise & Vibration

The effects of stratified charge combustion on full load and idle noise are discussed, together with vibration and torque fluctuations.

Topic Area (l) Cold Starting

This section records the effects of ambient temperature levels on cold starting and driveability, together with vehicle transient effects on the combustion system.

The actual search for published literature was conducted by inspection of the indexes of various libraries, and the retrieval of those sections which applied to stratified charge engines. The search went back to literature published in 1900, although the earliest references were from 1920. A list of 127 publications from which the references were taken is shown in appendix A of this report.

Theory and Definition of Stratified Charge Engines and Derivation of Baseline Curves for Gasoline and Diesel Engines for Comparison Purposes

INTRODUCTION

Before embarking on a study of the literature on stratified charge engines, some basis must be established for the scope of the study and examples of existing engines must be chosen, so that the advantages and disadvantages of these new engines can be clearly demonstrated. In the following sections the term stratified charge is defined and baselines for exhaust emissions and fuel consumption of gasoline and IDI diesel engines are established. Throughout this report, both gasoline and IDI diesel engine data is used widely for comparative purposes. The U.S. automotive industry is generally less familiar with the IDI diesel than the European industry, and the adoption of the diesel as one basis of comparison may be confusing for some readers. Nevertheless, many of the engines reviewed bore a strong resemblance to DI and IDI diesels and it was useful to make this comparison rather than to contrast these with the gasoline engine. At times within this report, the IDI diesel may look particularly attractive with regard to certain features and characteristics of the engine. However, it is not the object of this report to support the use of the diesel engine as an automotive powerplant, but only to use it as a basis for comparison where necessary.

THEORY

The thermal efficiency of the combustion processes in engines is remarkably efficient, and often exceeds 50%, Fig. A1. The efficiency of the cycle is governed by two principal parameters, the compression ratio and the mixture strength, i.e. the ratio of fuel to air in the cylinder. The unit of measurement is equivalence ratio where 1 represents stoichiometric conditions (called Ø on the figures).

The method of controlling the power output of an engine has a very important influence on the efficiency of the engine. In the diesel engine, the amount of air drawn into the cylinder during each stroke varies only slightly over a wide range of loads and speeds. To lower the engine load, the fuel flow is reduced. Therefore, as the load is reduced the ratio of fuel to air in the cylinder decreases, and the cycle becomes more efficient. On the other hand, the power produced by the spark ignited engine is also governed by the fuel flow, but the equivalence ratio cannot be reduced below a certain minimum level, or else the spark will not ignite the mixture. Therefore, the power produced by the spark ignited engine is governed by throttling the air entering the engine so that the ratio of fuel to air is retained within a certain tolerance band independent of the engine conditions. A direct result of this is that the thermal efficiency of the spark ignition engine is roughly independent of load and does not improve at low loads as does the unthrottled diesel engine combustion process. Therefore, unthrottled engines are inherently more efficient at light loads.

The reason for this improvement in efficiency at low equivalence ratios can be easily explained on the basis of the Otto-cycle formula, efficiency $1 - (\frac{1}{r})^{\gamma-1}$ The ratio of specific heats of the working fluid is a primary factor influencing the cycle efficiency, Fig. A2. The figure shows that for a burnt mixture of air and gasoline, the ratio of specific heats increases as the equivalence ratio is reduced. In addition, the gas temperatures which exist after burning a very lean mixture are much lower than for a stoichiometric mixture. For example, the maximum gas temperature in a spark ignition engine at the end of combustion is around 2500°K. On the other hand, in an unthrottled engine at light load, the maximum gas temperature may be only 1250°K. Both of these effects lead to a higher ratio of specific heats for the unthrottled engine at light load, and therefore better efficiency.

However, the thermal efficiency of the combustion process is not the complete story. The work expended by the cycle must be converted to rotating mechanical work through the engine structure. The engine itself absorbs some of this work in friction between the moving parts, such as the piston and liner, bearings and camshaft and valve gear. These parasitic losses of the mechanical components of the engine are closely related to the compression ratio of the engine, and the maximum cylinder pressures during the cycle. As the compression ratio increases, so does the friction. Spark ignition engines operate at low compression ratios, around 9, while compression ignition engines of the indirect injection type operate around 20.

Fig. A3 shows the energy balance for a typical gasoline and indirect injection diesel engine, as well as a hypothetical unthrottled engine with a moderate compression ratio. The appropriate engine sizes have been chosen so that the total work produced by each engine is the same. The diesel engine, of course, produces less power for each unit of swept volume due to the onset of exhaust smoke at high loads. Referring first to the upper portion of the figure, which shows an energy balance at full load for each engine, all having the same fuel flow. The diesel engine is the most efficient, but the mechanical friction caused by the high compression ratio reduces the output work to the same level as the gasoline engine.The hypothetical engine also has the same output. Part of the rejected heat and the friction appears as heat in the coolant and oil, while the remainder is rejected in the exhaust gases. In the gasoline engine there is an additional energy loss caused by the rich mixture, i.e. excess fuel in the cylinder beyond that required for stoichiometric combustion. This appears in the exhaust as carbon monoxide which still has a calorific value and therefore contains chemical energy.

The net effect at full load is that the overall efficiency of all these types of engines is identical. However, when the engine is operating at low loads the position changes radically (lower half of Fig. A3). The diesel engine combustion process is considerably more efficient than the gasoline engine, but the energy expended in friction remains the same as at full load, in absolute terms, and therefore is proportionally greater at part load. The end result is that a portion of the thermal efficiency advantage of the unthrottled diesel combustion process is lost in friction, and the overall gains over the throttled gasoline engine are relatively small. The friction of the gasoline engine is now split into two components, one mechanical friction and the other a pumping term caused by drawing the air past the throttle in the intake. In contrast, the hypothetical unthrottled engine operating at a compression ratio of 12 with relatively good thermal efficiency and moderate friction has the highest work output, and therefore the highest overall efficiency, between fuel entering the engine and work at the engine output shaft.

Therefore, a combustion process which operates at a compression ratio of about 12 will give the best fuel consumption for light load duties such as motor cars. However, there is no such combustion process. Compression ignition engines will not start at this compression ratio and spark ignition engines will detonate under these conditions. The answer is to amalgamate the spark ignition and unthrottled diesel combustion processes into a hybrid form called stratified charge. In its simplest form, fuel is injected into the centre of swirling air in the combustion chamber at some time before the end of the compression stroke, Fig. A4. The injection timing is chosen such that the fuel has sufficient time to evaporate and diffuse into the air, so that an ignitible mixture with an equivalence ratio between about 0.8 and 1.4 occurs at the spark plug near TDC. The mixture is then ignited and the fuel is burnt at an overall mixture strength which the spark plug could not ignite if the mixture were homogeneous.

DEFINITION AND CLASSIFICATION OF STRATIFIED CHARGE ENGINES

Definition: A stratified charge engine may be defined as an engine with intermittent combustion initiated by a spark plug, where the element of mixture ignited by the spark plug is not typical of the mixture in the remainder of the working gas, over some portion of the engine operating regime.

Before ignition, the remainder of the working gas will correspond to one of two cases:

(a) a mixture of air and fuel whose average mixture strength is leaner than that ignited by the plug. In addition, there may or may not be exhaust residuals from a previous working cycle distributed homogeneously throughout the working gas.

(b) a mixture composed of the same ratio of air and fuel as that at the spark plug, but with considerable dilution by exhaust residuals from a previous working cycle.

In a review of this kind it would be erroneous and confusing to consider all stratified charge engines together. In fact, the term stratified charge encompasses a wide range of engines lying between the general class of throttled gasoline and unthrottled diesel engines. Furthermore, the characteristics of different types of stratified charge engines can differ, as much as gasoline engines differ from diesels. To simplify presentation, the various engine types were divided into the minimum number of different categories. The categories were chosen so that all the engines within a single category had a similar combustion chamber configuration and nature of combustion.

Category 1

Single combustion chamber with fuel injection before combustion, mixture formation by fuel injection and/or air movement.

Category 2

Single combustion chamber with fuel injection during combustion and combustion controlled by injection. This is really a special case of category 1 but since it encompasses one of the principal stratified charge developments, i.e. TCCS, it was convenient to consider the process in a special category. In addition, the characteristics of this category differ from those of category 1.

Category 3

Single combustion chamber with fuel injection on to the piston. Combustion controlled by fuel evaporation from the piston.

Category 4

Split combustion chambers with fuel injection into a prechamber. In some cases fuel is also added to the induced air.

Category 5

Split combustion chambers, with a separate inlet valve for pre-chamber.

Category 6

Other weak mixture spark ignition engines, based on carburettors.

Category 7

Two strokes, exhaust diluent engines and miscellaneous.

Some examples of each category are shown in Table A1.

DERIVATION OF BASELINE DATA

Fuel Economy and Exhaust Emission

The final test of any automotive engine with respect to exhaust emissions, is the performance in the CVS-CH tests. Generally, the fuel economy of the engine is also calculated from a carbon balance of the engine exhaust from this test. This is generally accepted as a realistic measurement of the vehicle economy under city driving conditions, since it encompasses both hot and cold starting conditions. It was particularly convenient to adopt the CVS cycle for fuel economy measurements of stratified charge engines as this information has been published on many of the engines reviewed. Before the stratified charge engine can be compared with the existing gasoline engine, a baseline fuel economy must be established for that gasoline engine. This is notoriously difficult as many factors influence the fuel economy i.e. vehicle weight, gear ratios, transmission type, mixture strength settings, etc. The most important factor is vehicle weight, but engine mixture strength is also known to have a considerable effect. A method was derived to correct vehicle fuel economies of gasoline engines based on the mixture strength. The CO mass emission during the CVS test was adopted as a measure of the mixture strength. If the chemical energy contained in the CO is referred to an appropriate amount of engine fuel, then the fuel consumption can be reduced or increased accordingly. Therefore, for any given CVS CH test result, the fuel consumption figure at a certain CO level can be corrected to another CO emission level. This hypothesis was checked using two European vehicles; one fitted with a carburettor Car 1, and the other with fuel injection Car 2. A number of CVS CH tests were conducted with each car, with different mixture settings of carburettor and injection. The results in terms of fuel consumption against CO emissions followed a straight line relationship very close to that predicted by the chemical energy of the CO, Fig. A5. Therefore, to compare a wide range of gasoline cars, the fuel consumption figures were corrected to 30 g/mile CO. This CO level represents an average figure for well tuned cars of reasonable driveability and without after-burning devices. Most of the results on cars below 1360 kg were recorded in Ricardo laboratories and the rest were from 1974 certification tests published by EPA. The corrected fuel consumptions are plotted against vehicle inertia weight and refer to cars with sum weighted NO_x emissions in the range of 2 - 4.5 g/mile. Fig. A6. There is a well defined maximum fuel economy for each inertia weight and a line can be drawn through them, only missing two unrepresentative points. This line then gives a fuel economy value for well tuned gasoline engines without stringent NO_x control. This data all refers to vehicles with manual transmissions, and some loss would be expected if automatic transmissions were fitted. Ref. G40 suggests that this fuel economy loss would be about 1.6 m per U.S. gallon for an intermediate size vehicle.

Table Al. STRATIFIED CHARGE ENGINE CATEGORIES

1	2	3	4	5	6	7
SINGLE CHAMBERS			DUAL CHAMBERS		SINGLE CHAMBER	
FUEL INJECTION				CARBURETTORS		
Early Injection Into moving air	Late Injection Into moving air	Combustion Controlled by wall evaporation	Fuel Injection into prechamber also fuel addition to main chamber	Mixture supplied by carburettor(s) separate prechamber inlet valve		Two strokes, exhaust diluent engines Miscellaneous
EXAMPLES						
Ford Proco FCP Mitsubishi Hesselman Witsky	Texaco TCCS Curtiss-Wright Deutz	Man F-M	Newhall VW Huber Porsche Broderson	Honda CVCC Nilov Heintz GM Ford Nissan Walker	I.F.P.	Ricardo Jessel Kushul Nice YOCP

An unusual attitude adopted in this report has been to choose the best gasoline engines, rather than average engines, as the baseline against which the new stratified charge engines should be compared. The justification for this approach was that an automobile manufacturer would usually prefer to improve his product to the level of the best current engine, rather than adopting a new stratified charge engine which could mean new production equipment for cylinder heads and fuelling equipment.

Inspection of Baseline Gasoline Engine Curve

The baseline curve was inspected to find any anomalies in the data. The curve was re-drawn on the basis of fuel consumption per unit of vehicle weight, and there was seen to be a minimum point, Fig. A7. The increase in specific fuel consumption with smaller vehicles was expected, but the rising characteristic at high vehicle inertias could not be justified. In view of this, the curve was compared with a similar study undertaken by Ref. G39 and again there was an anomaly at the higher vehicle inertias, Fig. A8. The reason for the difference may be due to the restriction imposed in this survey to include manual transmission vehicles alone in the baseline curve and there were insufficient examples of good manual transmission vehicles at the higher inertia weights. In view of this, the baseline curve was modified to come into line with the results of Ref G39 at the higher inertias. Fig. A6, A7.

Other Baseline Curves

Two further baselines were constructed, for gasoline engines at low NO_x levels and for diesels. Four vehicles which had been fitted with high flow EGR valves, to bring the NO_x level to around 0.9 g/mile, Fig. A9, gave a 25% loss in economy. Results from diesel engines were also plotted, Fig. A10, to give a third baseline.

Multifuel engines will usually give a better fuel economy result when operating on diesel fuel rather than gasoline, due to the established practice of selling fuel on the basis of volume. The specific gravity of diesel fuel is about 11% greater than gasoline. Therefore, the diesel engine/gasoline engine fuel economy comparison was replotted on the basis of fuel weight, which reduced the relative gain of the diesel engine, Fig. A11. The calorific value of gasoline is 3-4% greater than diesel so that on an energy basis, even this weight comparison will be slightly modified. However, the weight comparison was considered to be the most useful in comparing vehicle results with engine test bed rate of fuel consumption, which are normally plotted in terms of fuel weight per unit of power.

Vehicle Performance

It is important that any vehicle which gives good fuel economy on the CVS test should also be capable of following the required acceleration rates. Obviously it is possible for a vehicle with high gear ratios or a very small engine, to give a good fuel economy result, but be incapable of following the driving cycle. Therefore, it is necessary to quantify the wheel torque available.

The most demanding portion of the CVS test is the acceleration between 190 and 200 seconds. An instantaneous acceleration rate of 5.3 km/hr/s is required at a speed of 53 km/hr. The accelerating force available at the circumference of the rear wheels can be calculated, using quoted figures of maximum engine torque and a typical torque curve.

$$F \qquad \frac{\eta T}{M}$$

η = transmission efficiency (assumed to be 90%)

M = vehicle speed/engine rotation speed in vehicle gear corresponding to 53 km/hr

T = engine torque at engine speed $\frac{53}{M}$

The wheel force required to give the accelerating rate of 5.3 km/hr/s was calculated and is shown on Fig. A6, together with the force available, for a number of vehicles. In general, the calculated CVS torque requirement is an overestimation, as the peak acceleration rate only occurs instantaneously and therefore, in an actual test, the driver will anticipate the higher rate and begin the acceleration before this point.

The vehicles with diesel engines were analysed in the same manner, and as expected the output was lower, Fig. A10. In testing, it was observed that some of these vehicles were on the border line of being able to follow the maximum acceleration rate, while others failed.

An example of the use of the baseline graphs for presenting performance economy and emission results from a gasoline engine is illustrated in Fig. A12. The vehicle displayed good economy, average performance and emissions which were outside the target for this survey. The triangle joins the 3 components of emissions from the CVS test, while the circle marks the targets. In this survey, sufficient information was not always available to calculate the rear wheel torque.

Method of Comparing Test Bed Data

Many of the results from stratified charge engines reviewed in this survey did not include CVS tests, but only single or multi cylinder test bed data. It was therefore necessary to have some baseline curves against which the engine could be compared. Obviously we are seeking a stratified charge engine which, when installed in a vehicle, will yield exhaust emission levels below the target, without exhaust aftertreatment. Therefore, the required test bed emission levels, expressed in g/bhp hr must be established. Since no gasoline engine will meet the target emission levels without aftertreatment, the alternative was to choose a diesel engine which will meet the target. The engine emission levels necessary to meet the target were established in a computer simulation exercise recorded in Ref. G41. These are illustrated in Fig. A13 and the computer study gave a CVS mass emissions figure of 0.46 g/mile HC and 1.17 g/mile NO_x with the target vehicle. (CO emissions were not estimated). Therefore, for an engine to achieve the target emissions, its specific emissions should lie within the curves for the IDI diesel. For convenience, the fuel consumption characteristics for the diesel were also chosen as a baseline, since the diesel is known to give a good fuel economy. Furthermore, unthrottled stratified charge engines were expected to give similar shapes of fuel consumption curves to the diesel.

The test bed curves have been presented on the basis of power to give a peak power value of 96 kW (128 BHP). Obviously the engines reviewed in this survey have different power outputs and to simplify comparisons, the results from these stratified charge engines were scaled to the target power. In the case of single cylinder results, where the specific emissions and fuel consumptions are distorted by the high engine friction levels, the results were modified by applying correction factors based on the single cylinder and predicted multi-cylinder friction levels.

In order that the emissions and consumption baseline graph may be related to current gasoline levels, two reports (Ref. G33 and G34) covering baseline exhaust emissions and consumption of a number of large U.S. gasoline engines were studied and one, representing good results of fuel economy and exhaust emissions, was chosen (GM350). Using the engine torque curve, the appropriate swept volume was calculated to meet the calculated peak power value of 96 kW. The relationship between specific exhause emissions and power for the chosen gasoline engine was then adjusted to the peak power of the target engine with the new displacement. The resulting data for the gasoline engine is shown with the diesel baseline in Fig. A14

Load Factor of the CVS-CH Test

Since the LA.4 cycle is to form the basis for measuring fuel consumption of stratified charge engines, it is important that the characteristics of the cycle should be understood. For example, stratified charge and diesel engines are expected to give similar or worse fuel consumption values to the conventional gasoline engine at full load, but a significant reduction in fuel consumption should be apparent at light load. Therefore, a driving cycle where the engine is frequently required to operate near full load should show no advantage to unthrottled engines, while the true potential for light duty use would become obvious in a representative lightly loaded cycle. Therefore, a typical engine load and speed history in a CVS test should be known.

A computer program for simulating the LA-4 driving cycle was available, and this program determined the engine power requirements during each one second portion of the cycle. The power requirement was based on;

(a) An empirical expression for vehicle rolling resistance and air drag based on vehicle dimensions and weight.

(b) Input data on vehicle transmission ratios and efficiencies in each gear, and wheel diameter.

(c) Input data on the vehicle inertia.

(d) Input data on engine torque output and rotational inertia.

(e) Cycle speed and acceleration rates.

The program could also sum the total time spent by the engine at each point in the power spectrum. Two vehicles were subjected to this analysis:

(a) A 1590 kg (3500 lb) car with manual transmission, 3 speed gearbox, 4.17 litre (254 in^3) engine.

(b) A 1140 kg (2500 lb) car with manual transmission, 4 speed gearbox, 2.31 litre (141 in^3) engine.

The result of this calculation is shown in Figs. A15 and A16, where the time spent by the engine at low load and under power is identified. This information was further processed to obtain the proportion of the total fuel, consumed at each load and speed. A specific fuel consumption map from a typical stratified charge engine was used, and the results indicate that a large proportion (45%) of the fuel in the CVS test is consumed at specific loads between 0.7 bar (10 lbf/in^2) and 4.2 bar (60 lbf/in^2) BMEP, Figs. A17 and A18.

Other Driving Cycles

As a further test of the baseline curves for CVS-CH vehicle fuel economy, results from other driving cycles were inspected. Naturally, a difference in actual economy level was expected. However, some driving cycles will show up the fuel economy gains of stratified charge more so than others. Since the diesel engine is an example of an unthrottled engine, it was considered that the gains of the diesel engine over gasoline engines should be identified over a number of driving cycles.

Many automotive publications, in road tests on vehicles, present figures of fuel economy under city driving conditions, Fig. A19. The difference in fuel economy between the gasoline and the diesel engines is less than that of the CVS-CH cycle, Fig. A20. The ECE 15 driving cycle Fig. A22 is used for European emission regulation, and involves exceptionally low rates of vehicle acceleration. The advantage of the diesel engine is much less pronounced on this cycle, Figs. A20 and A21. Therefore, the conclusion of these comparisons with other driving cycles is that the CVS cycle emphasizes the gains of unthrottled engines.

Figure A1. Indicated thermal efficiency of reciprocating I.C. engines.

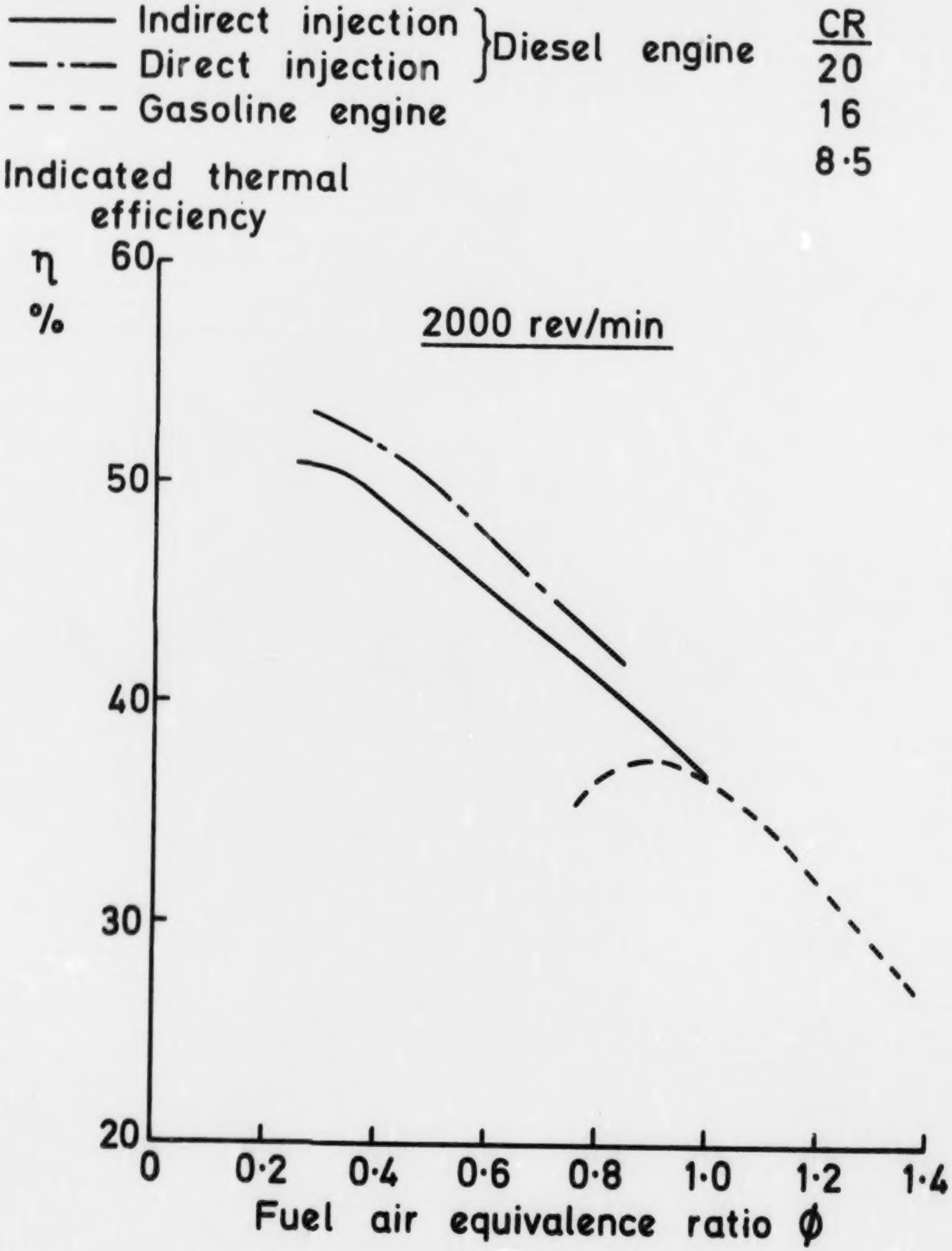


Figure A2. Ratio of specific heats for a burnt mixture of air and gasoline.

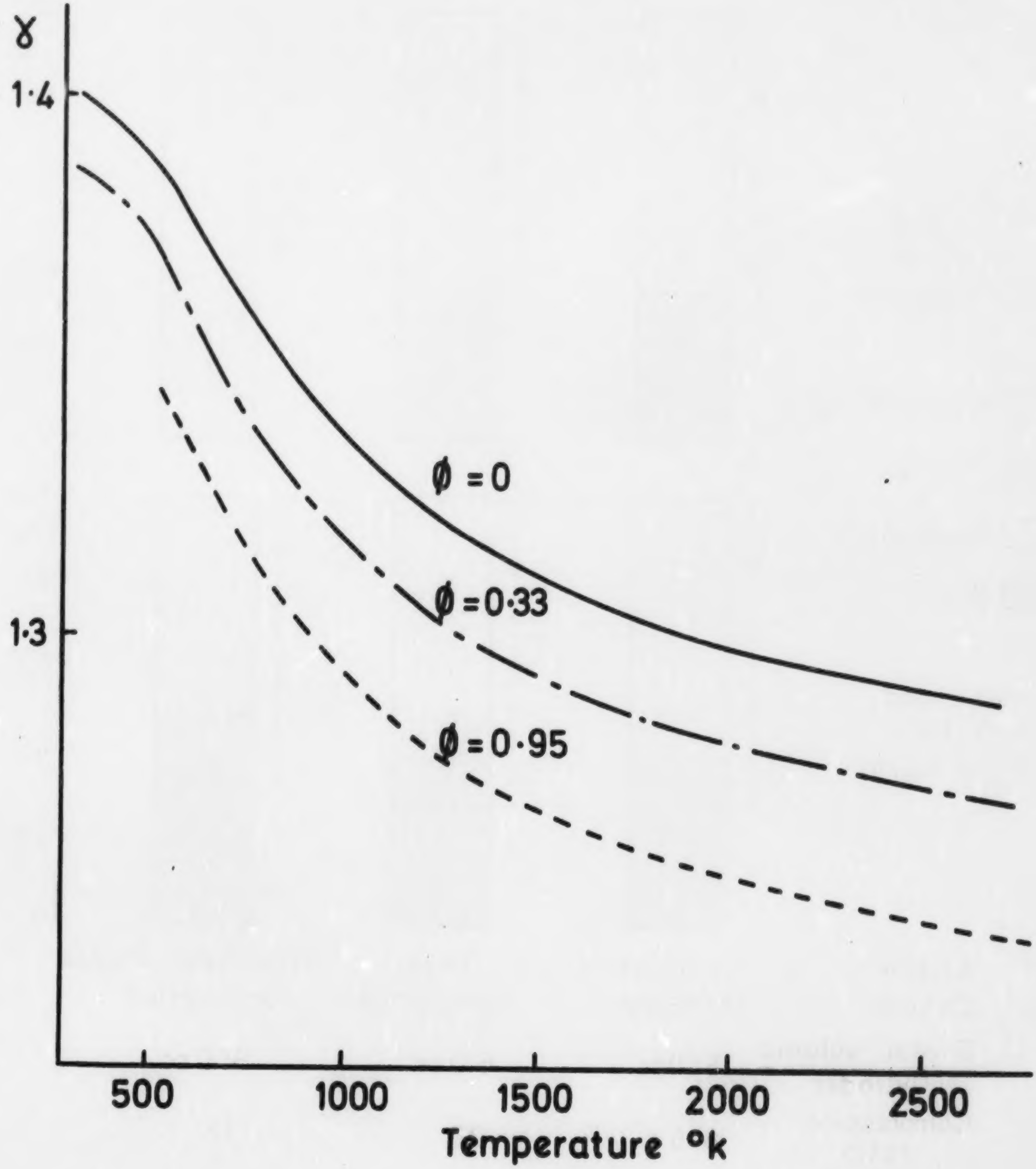


Figure A3. Energy balance for engines 2000 revolutions per minute.

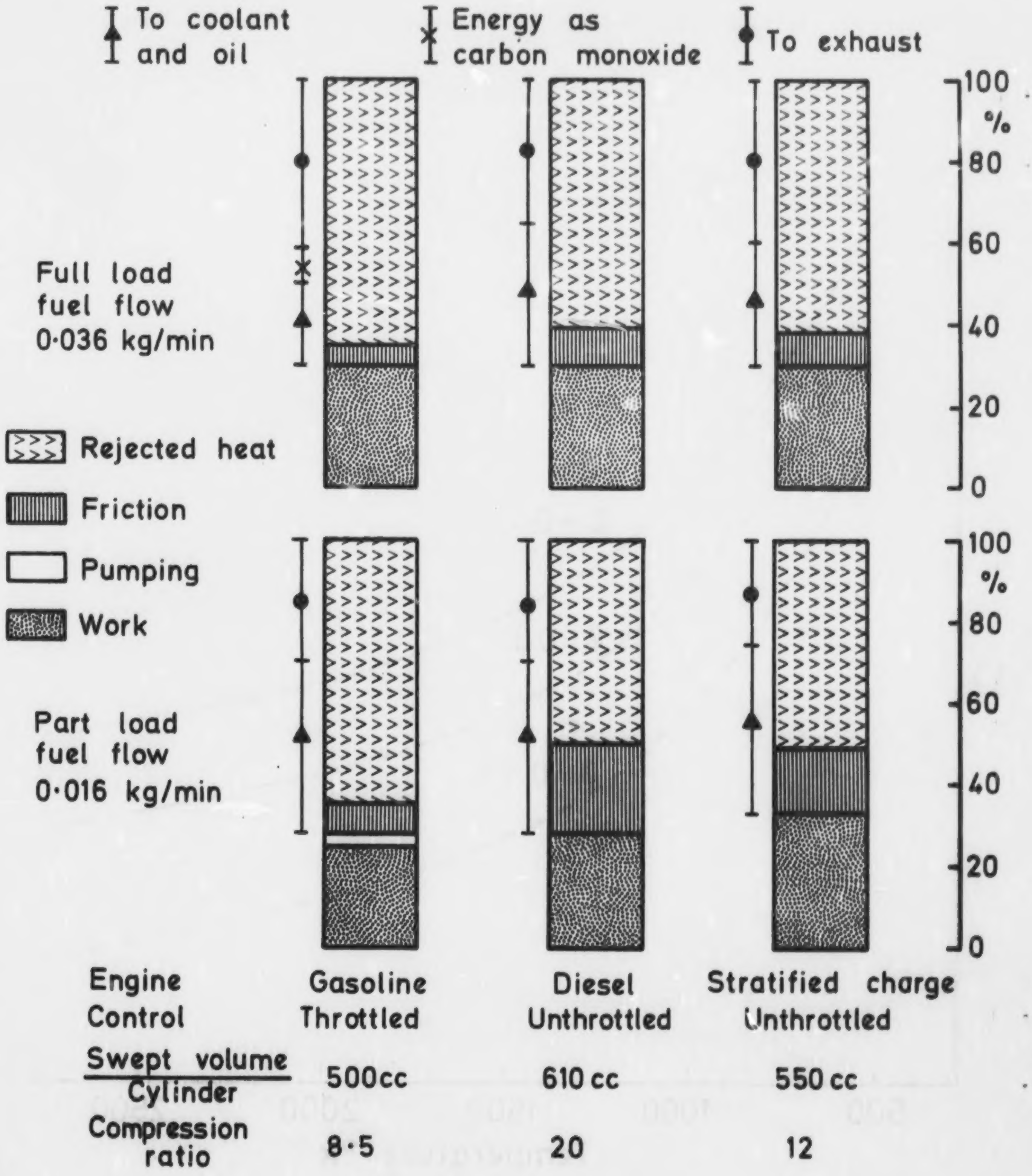


Figure A4. Unthrottled stratified charge engine.

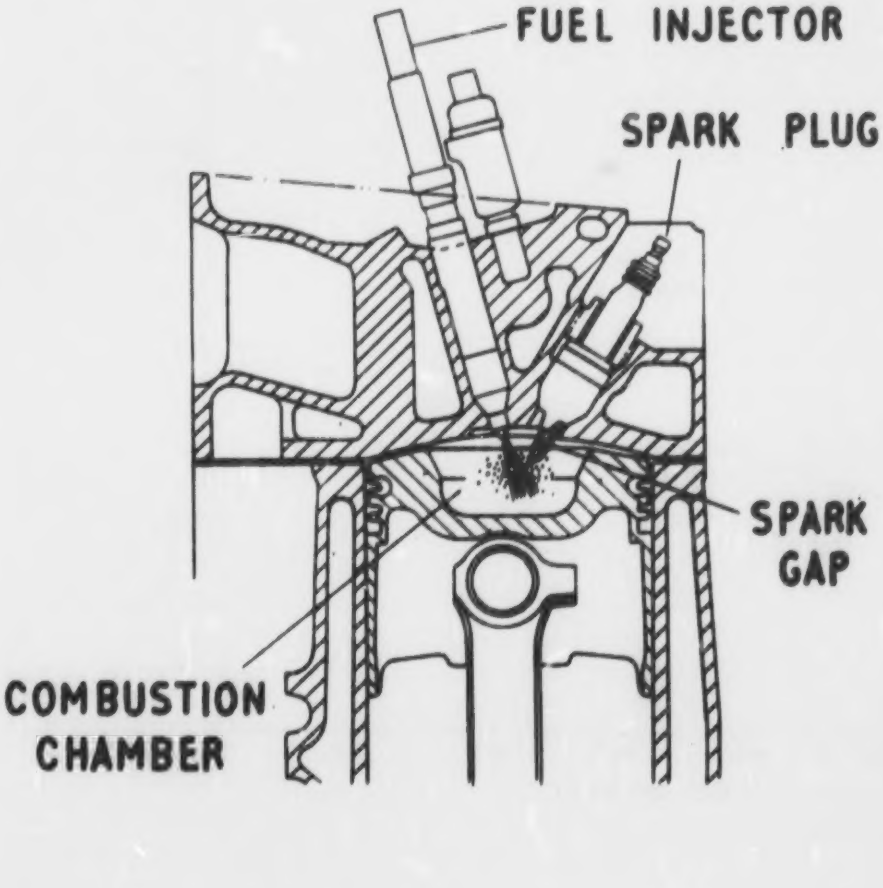

FUEL INJECTOR
SPARK PLUG
SPARK GAP
COMBUSTION CHAMBER

Figure A5. Relationship between fuel consumption and carbon monoxide in CVS-CH test. (no emissions control).

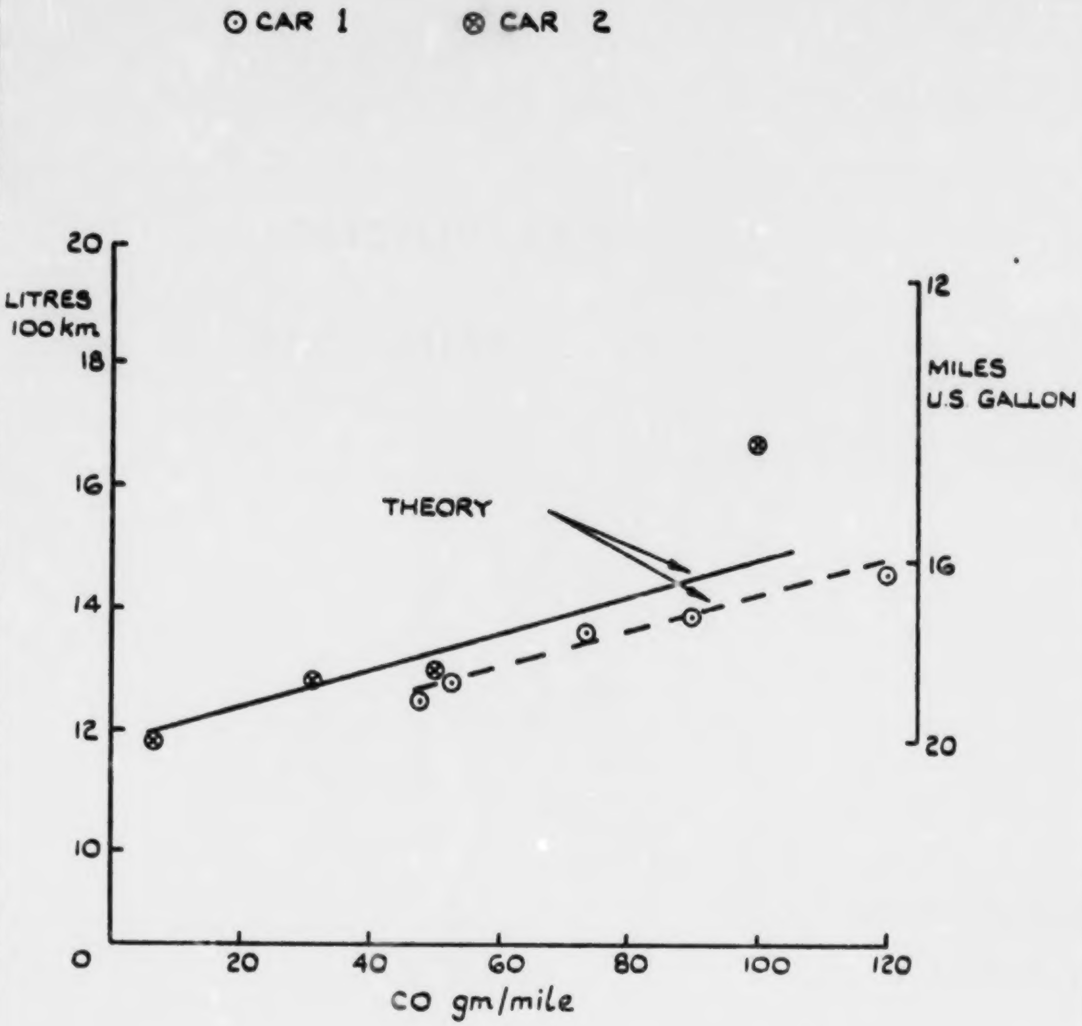


Figure A6. Fuel consumption of gasoline vehicles on CVS-CH test corrected to 30 gm/mile CO.

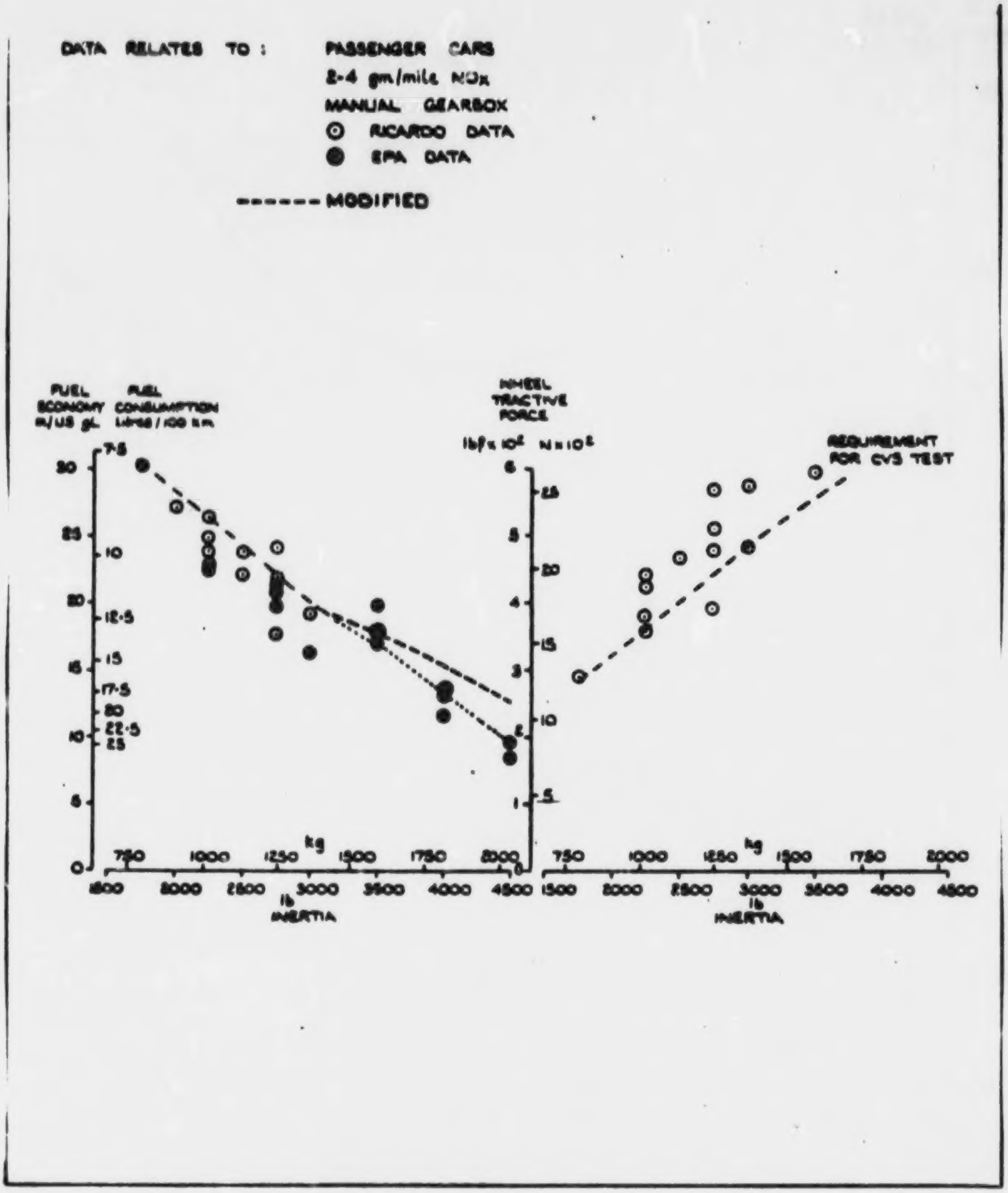


Figure A7. Baseline fuel economy curve for gasoline engine. Drawn on the basis of fuel consumption per unit of vehicle weight.

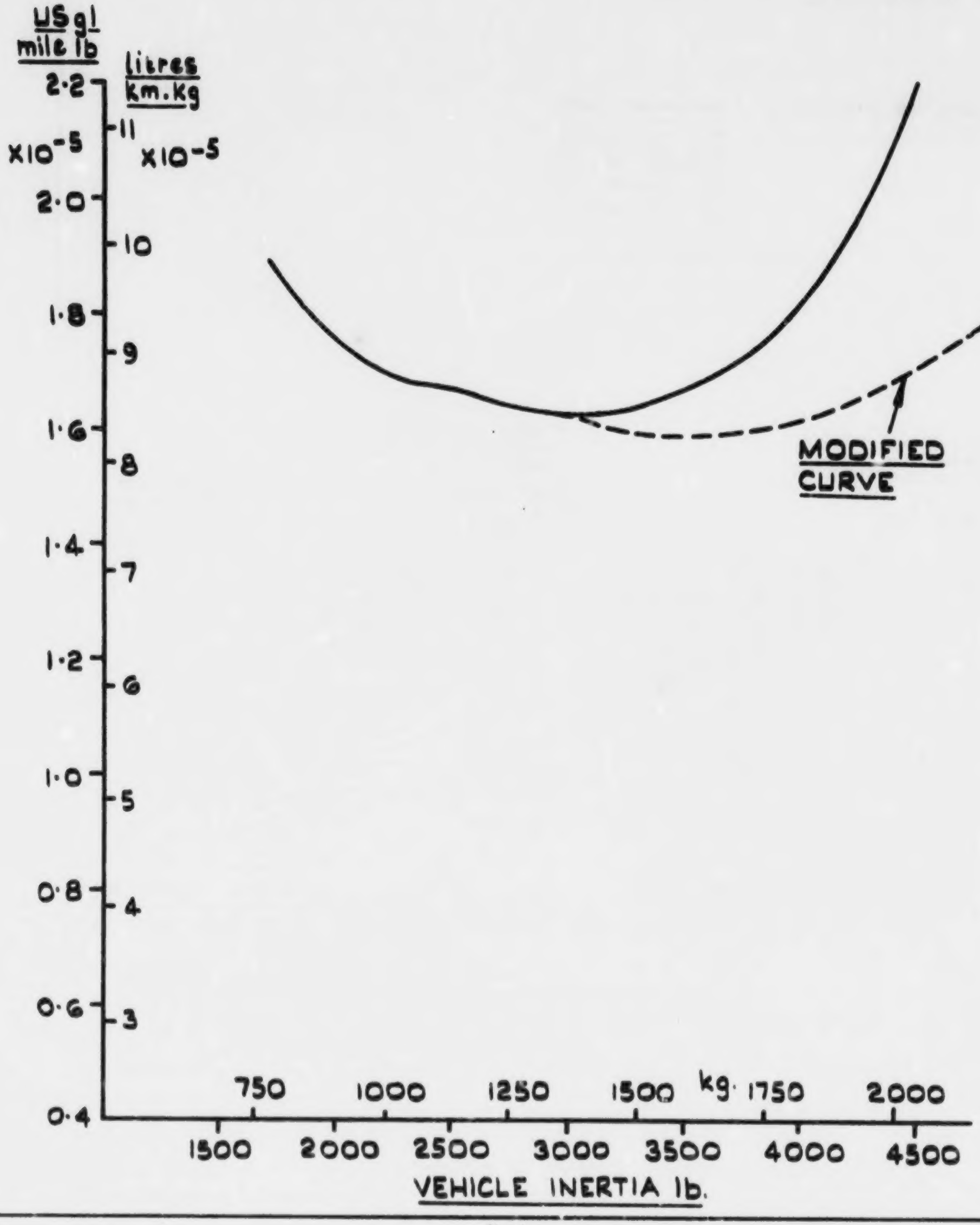


Figure A8. Comparison of fuel economy baseline for gasoline engines compared with similar baseline established by Brogan.

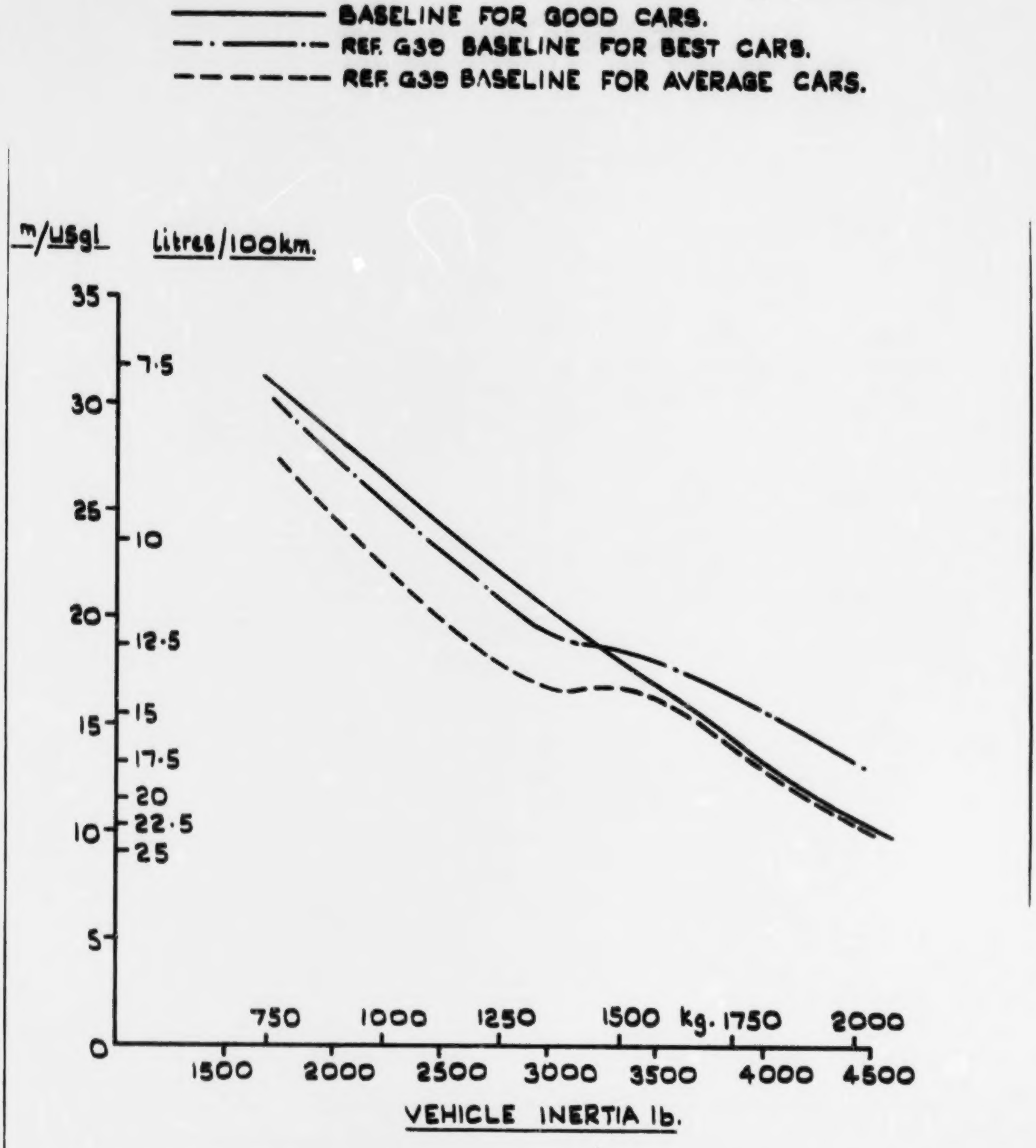


Figure A9. Fuel consumption of vehicles on CVS-CH test corrected to 30 gm/mile CO.

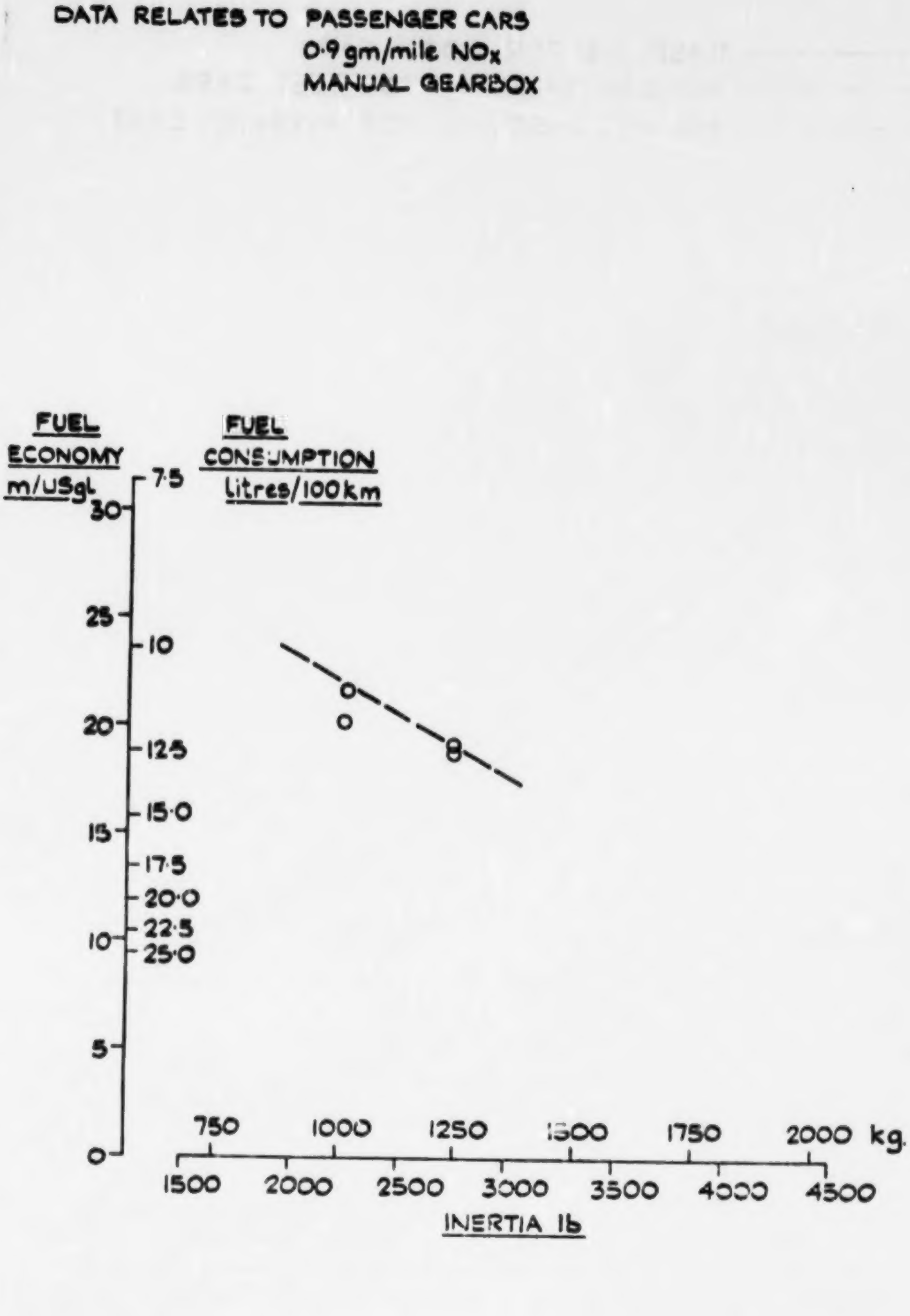


Figure A10. Fuel consumption and emissions on CVS-CH test of diesel cars compared with gasoline cars.

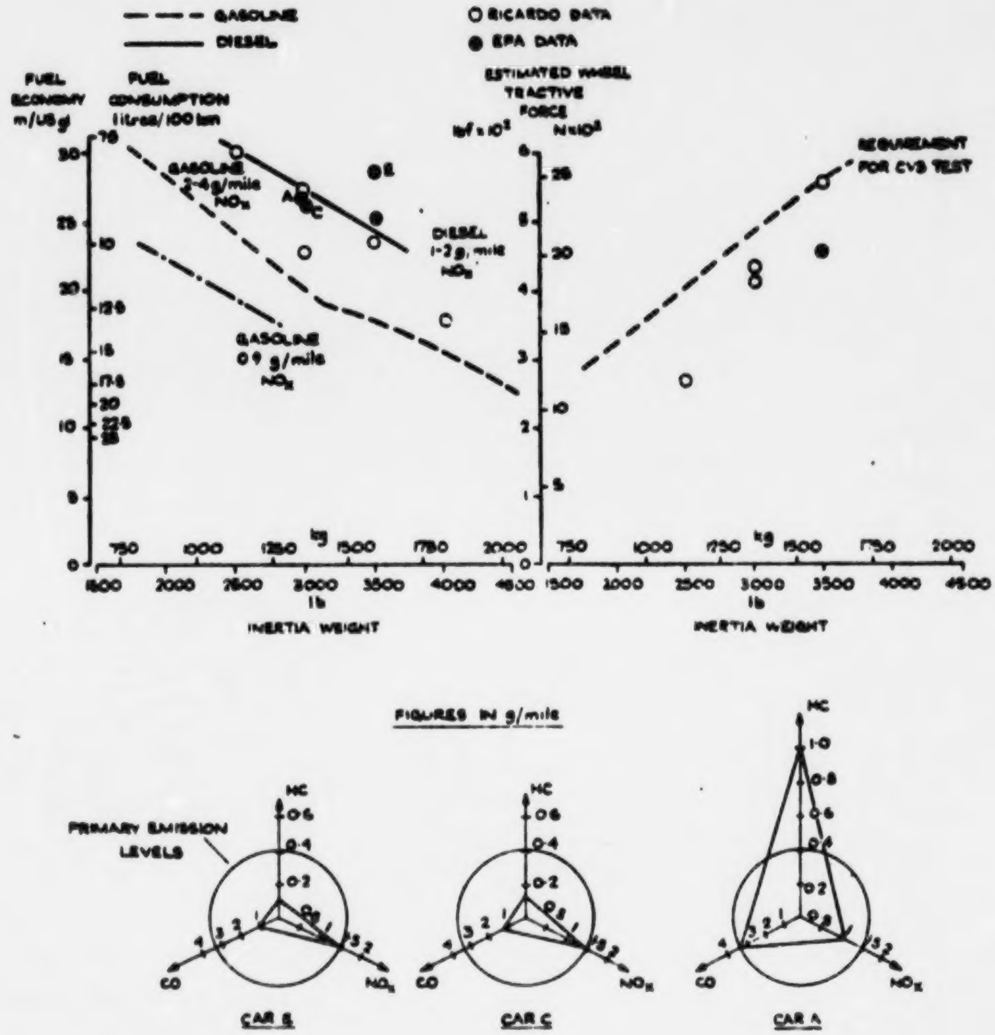


Figure A11. Fuel economy of diesel and gasoline vehicles on basis of fuel weight.

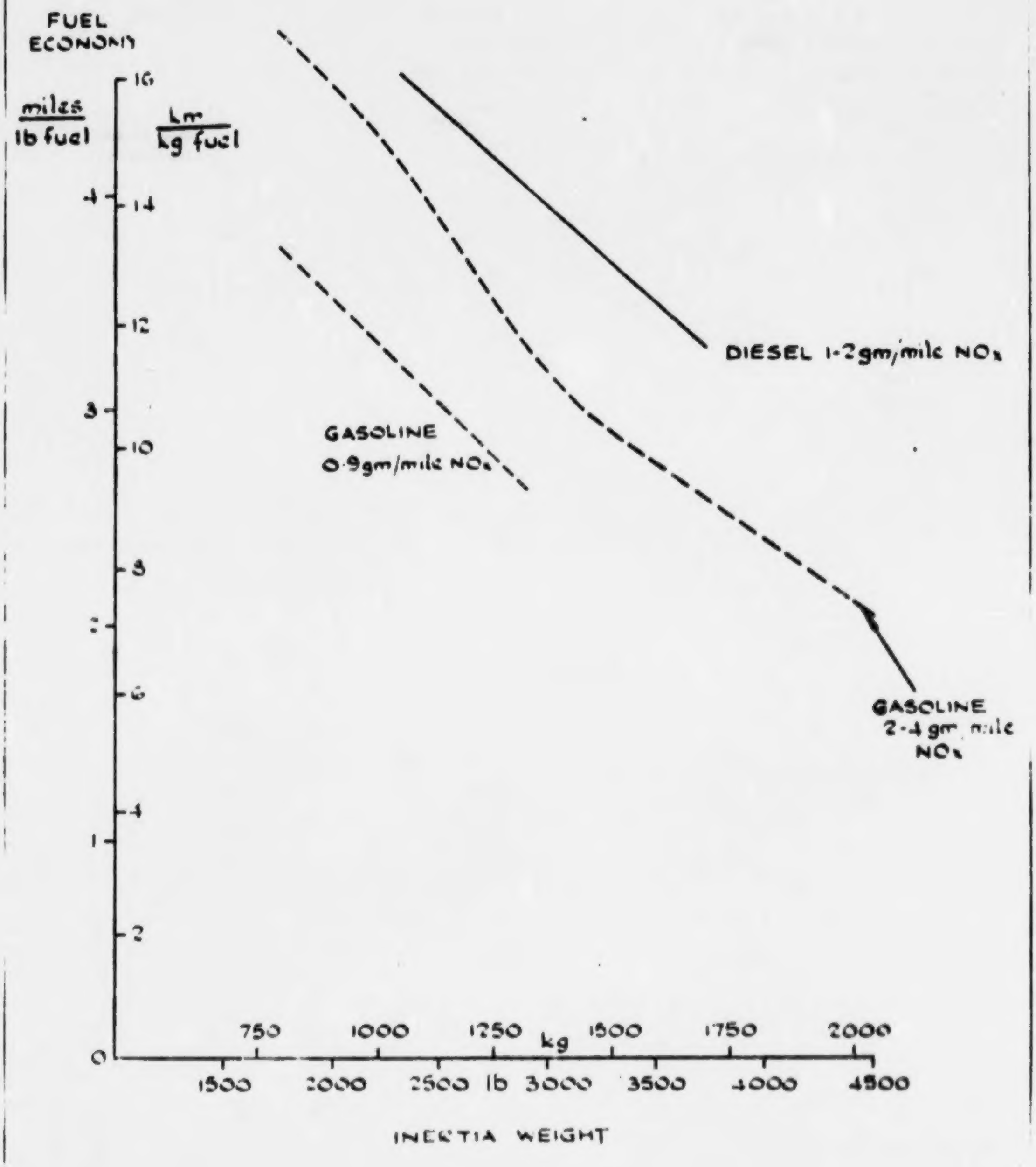


Figure A12. Example of presentation of CVS-CH test result of emissions and fuel economy together with the calculated maximum wheel tractive force in the test.

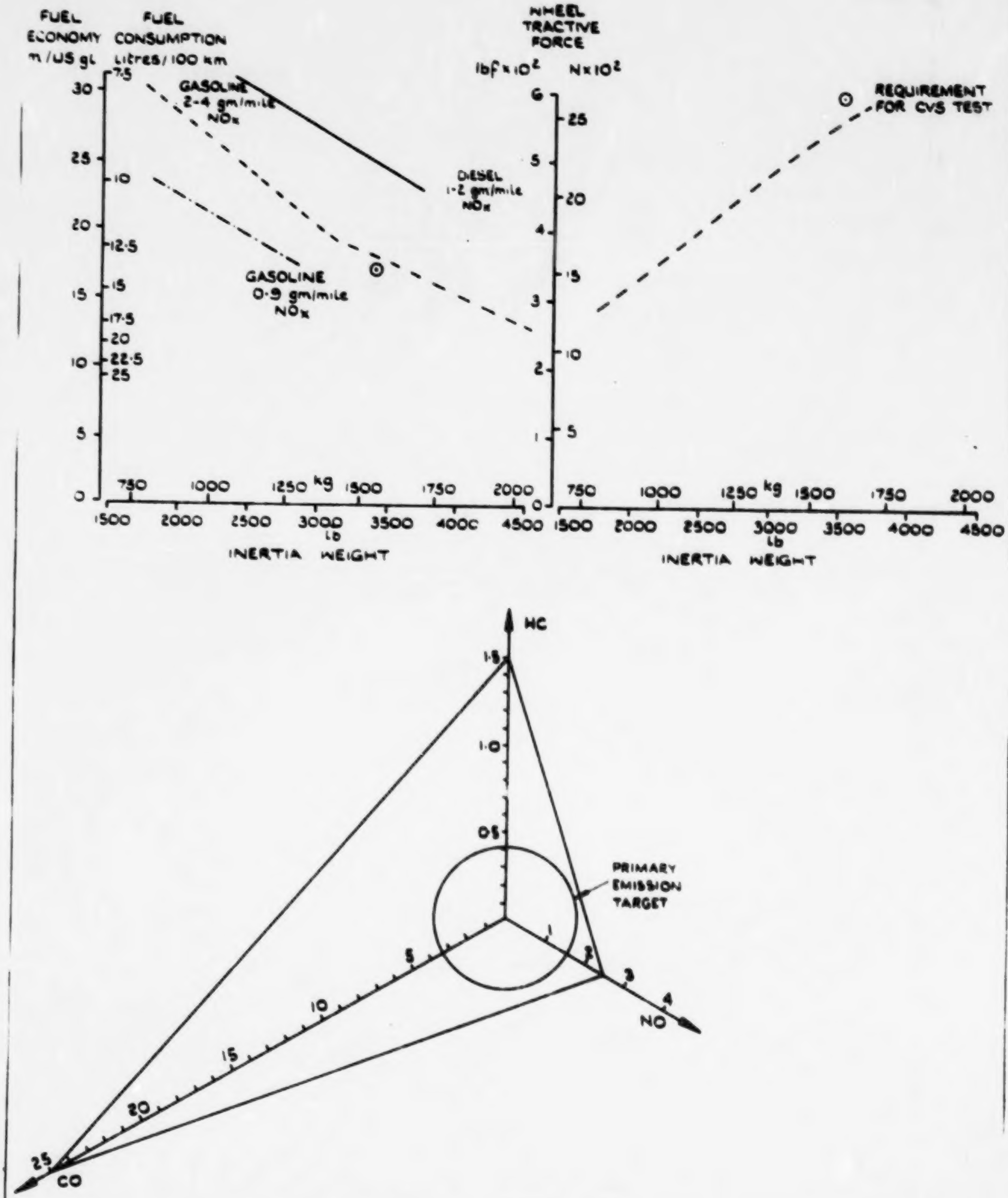


Figure A13. Emissions and fuel consumption of a Comet Mk V diesel engine to be used as a baseline.

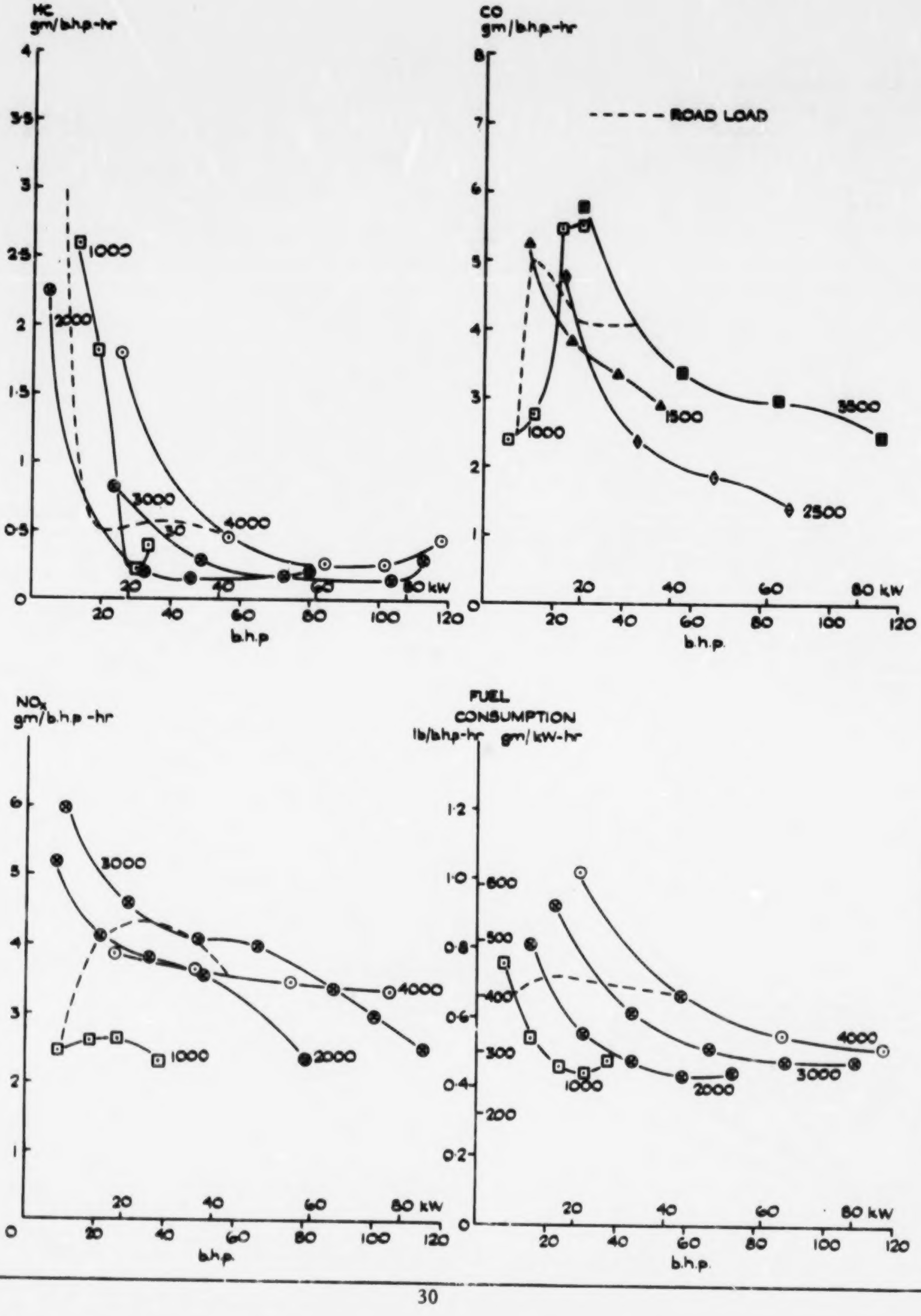


Figure A14. Comparison of emissions and fuel consumption of Comet Mk V diesel engine and gasoline engine (data scaled from 5·741 engine).

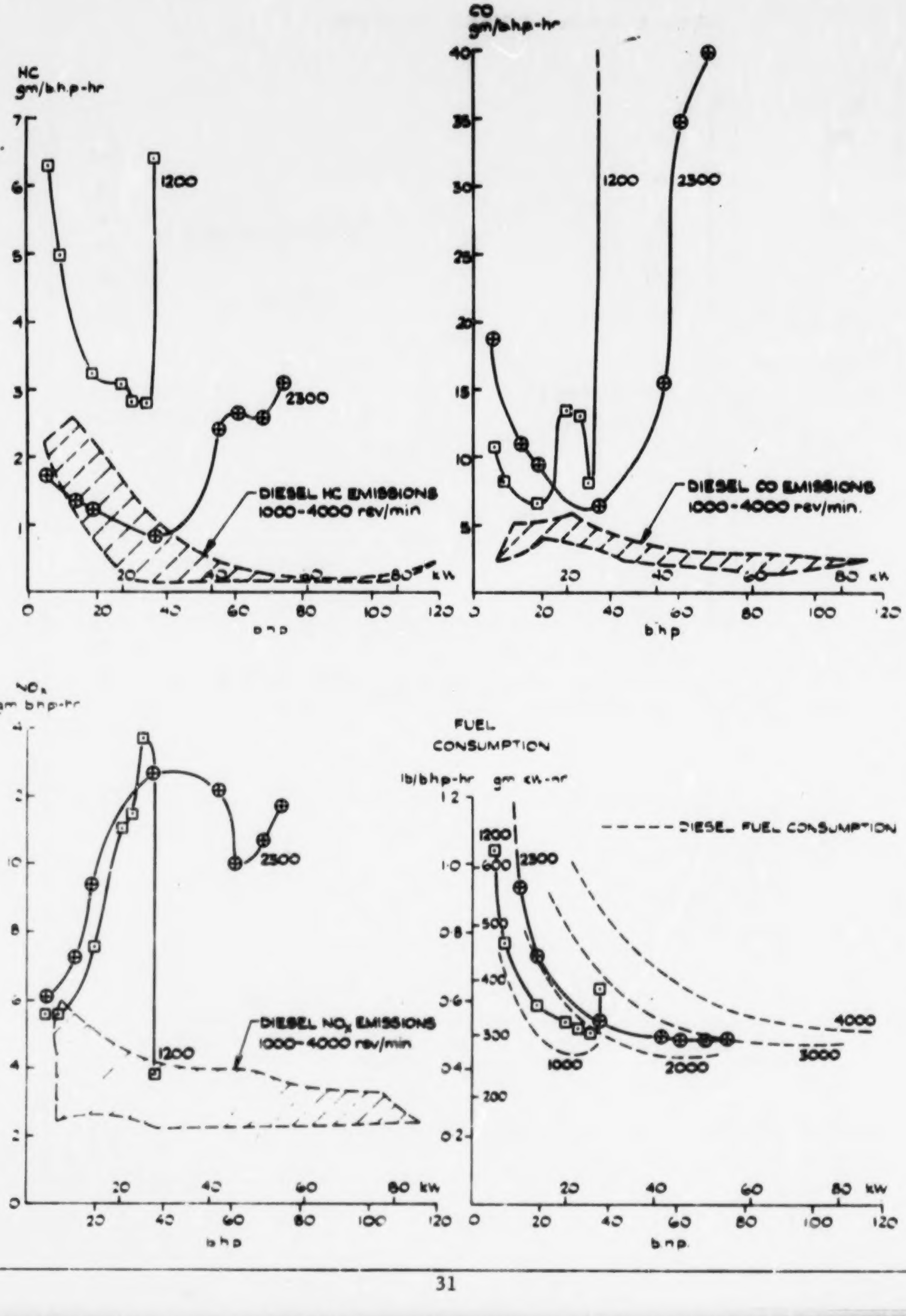


Figure A15. Time spent at each load and speed in CVS test.

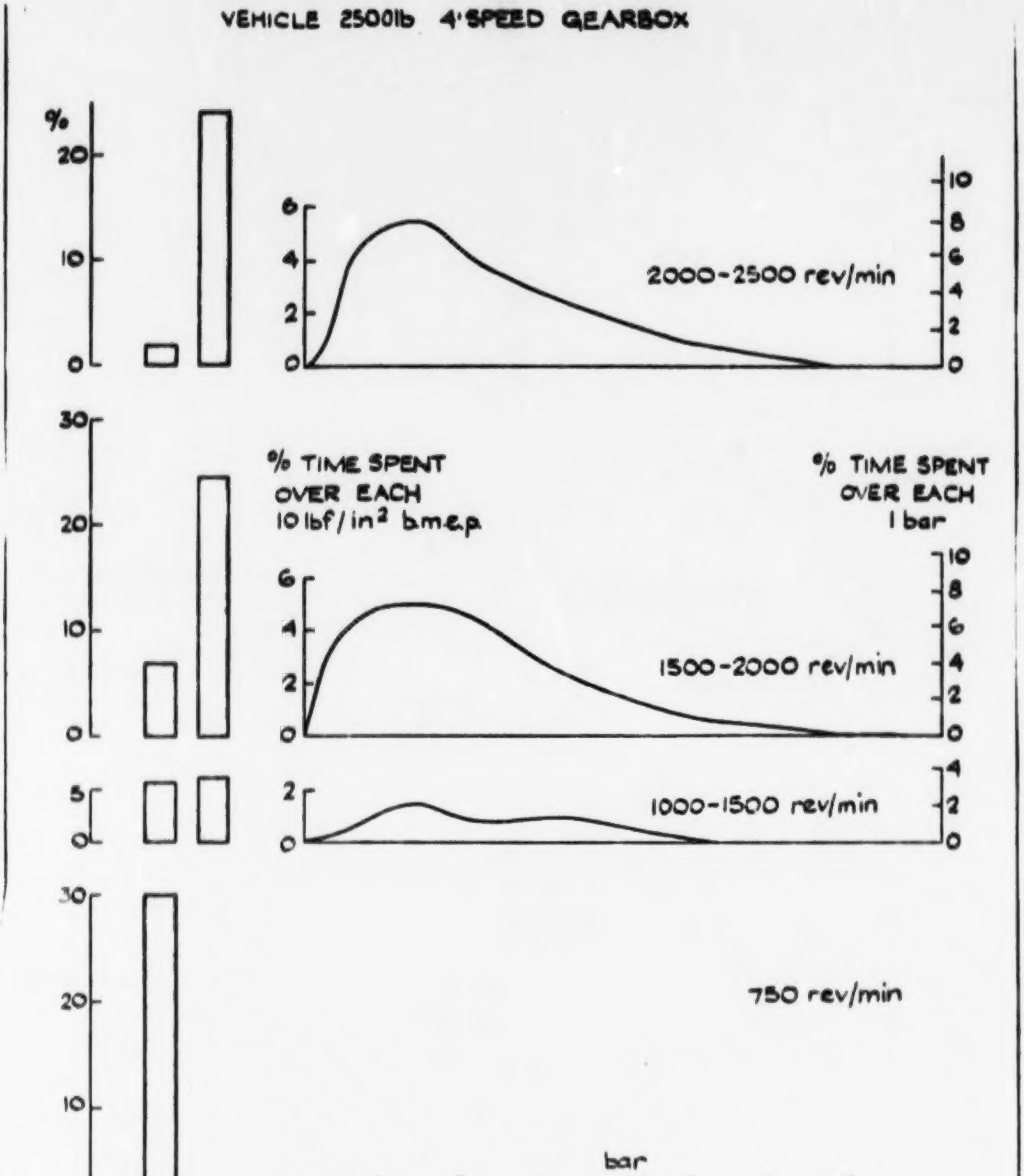


Figure A16. Time spent at each load and speed in CVS test.

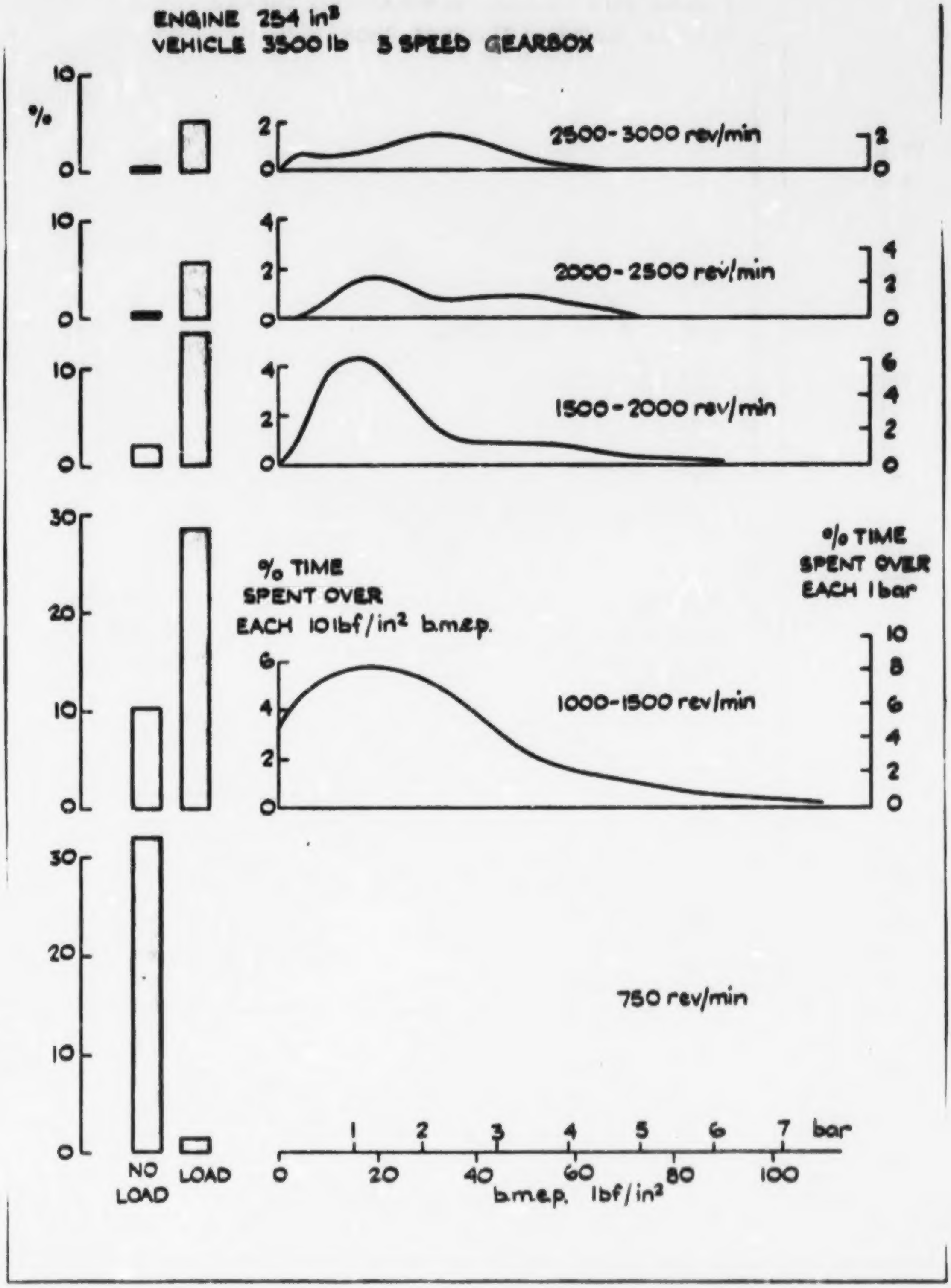


Figure A17. Amount of fuel consumed at each load and speed in CVS test.

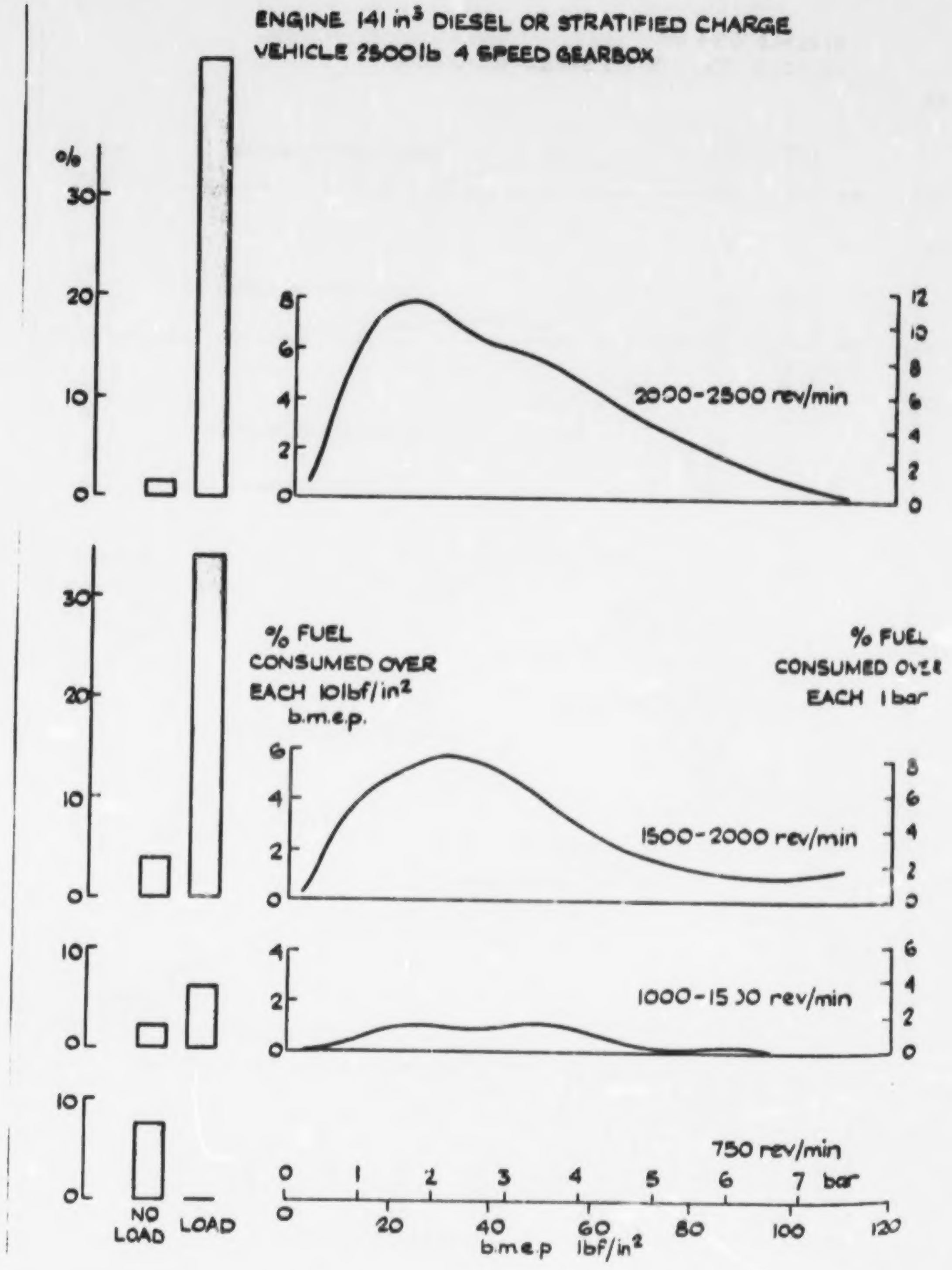


Figure A18. Amount of fuel consumed at each load and speed in CVS test.

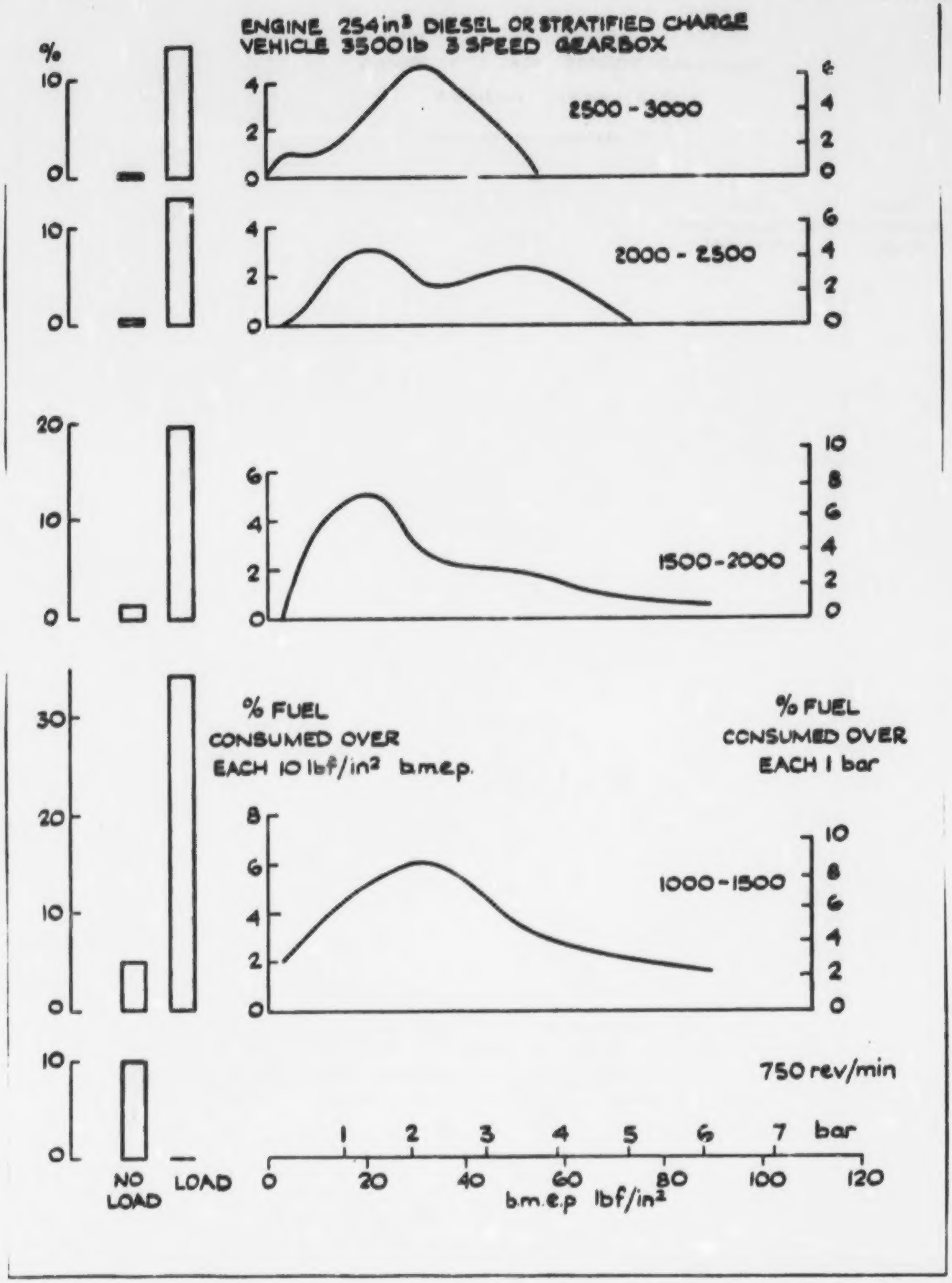


Figure A19. Quoted fuel economy figures for a variety of gasoline and diesel powered vehicles.

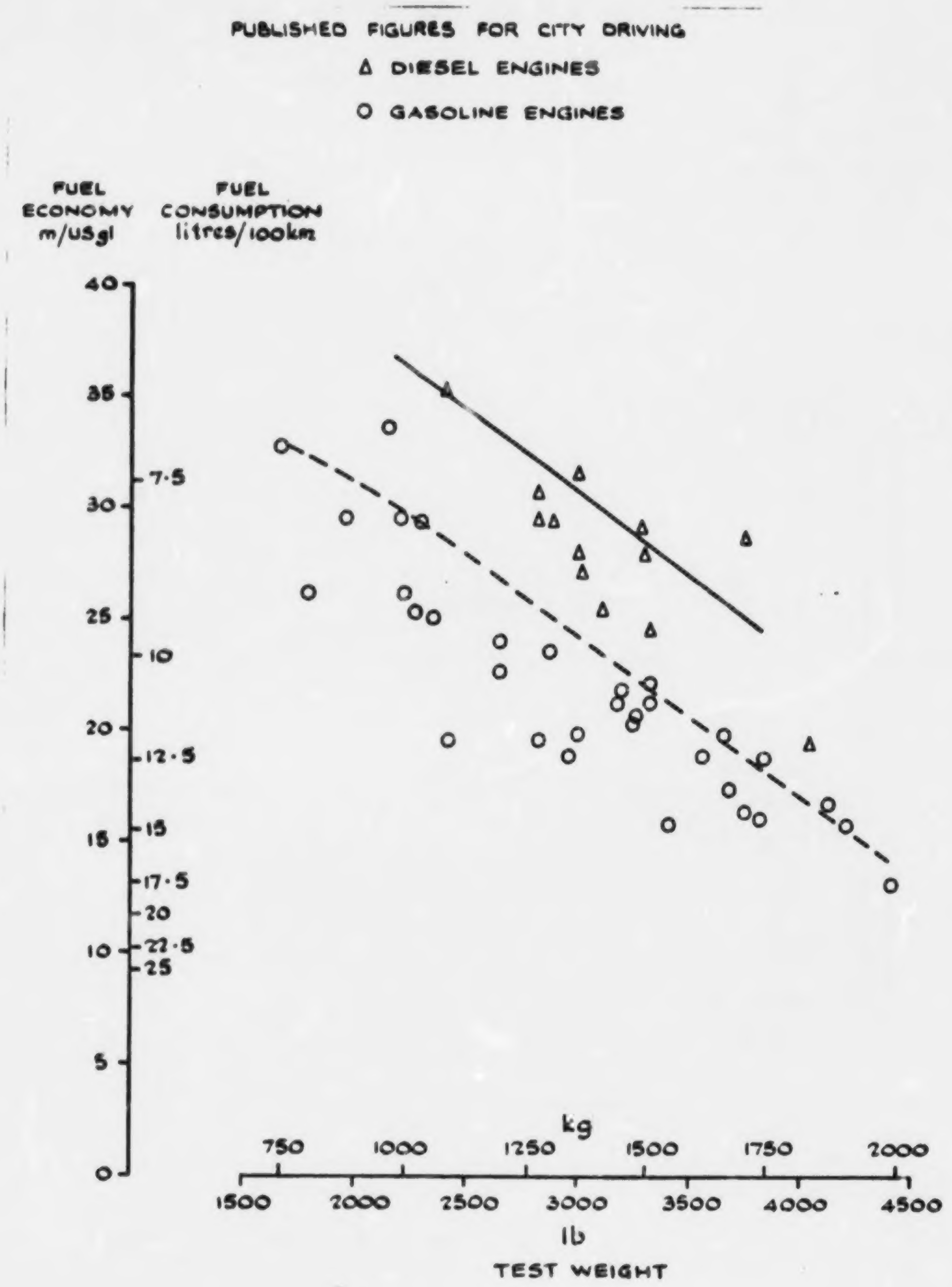


Figure A20. Fuel economy gain of diesel over gasoline engine.

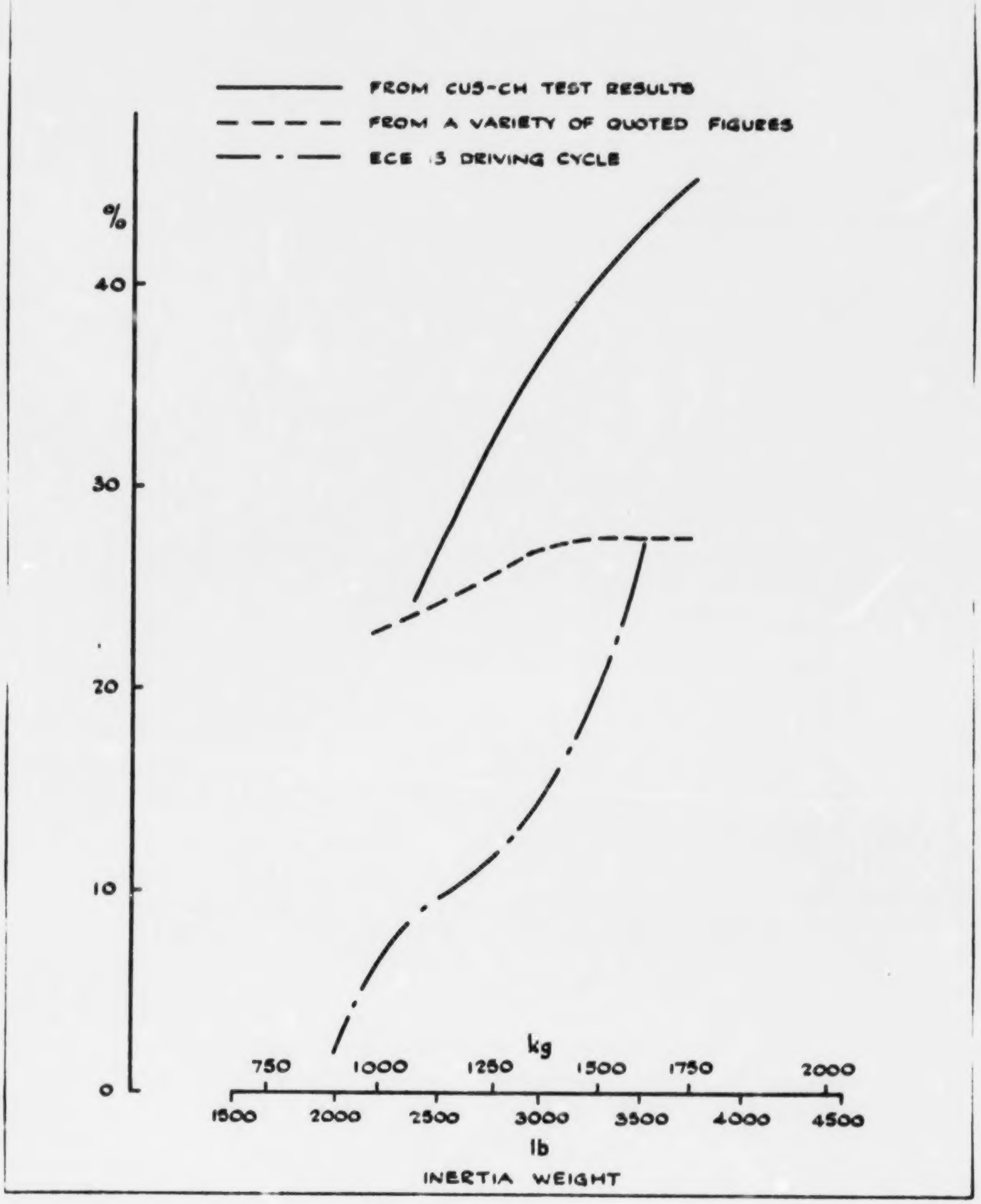


Figure A21. Comparison of fuel consumption of diesel and gasoline vehicles on ECE-15 driving cycle.

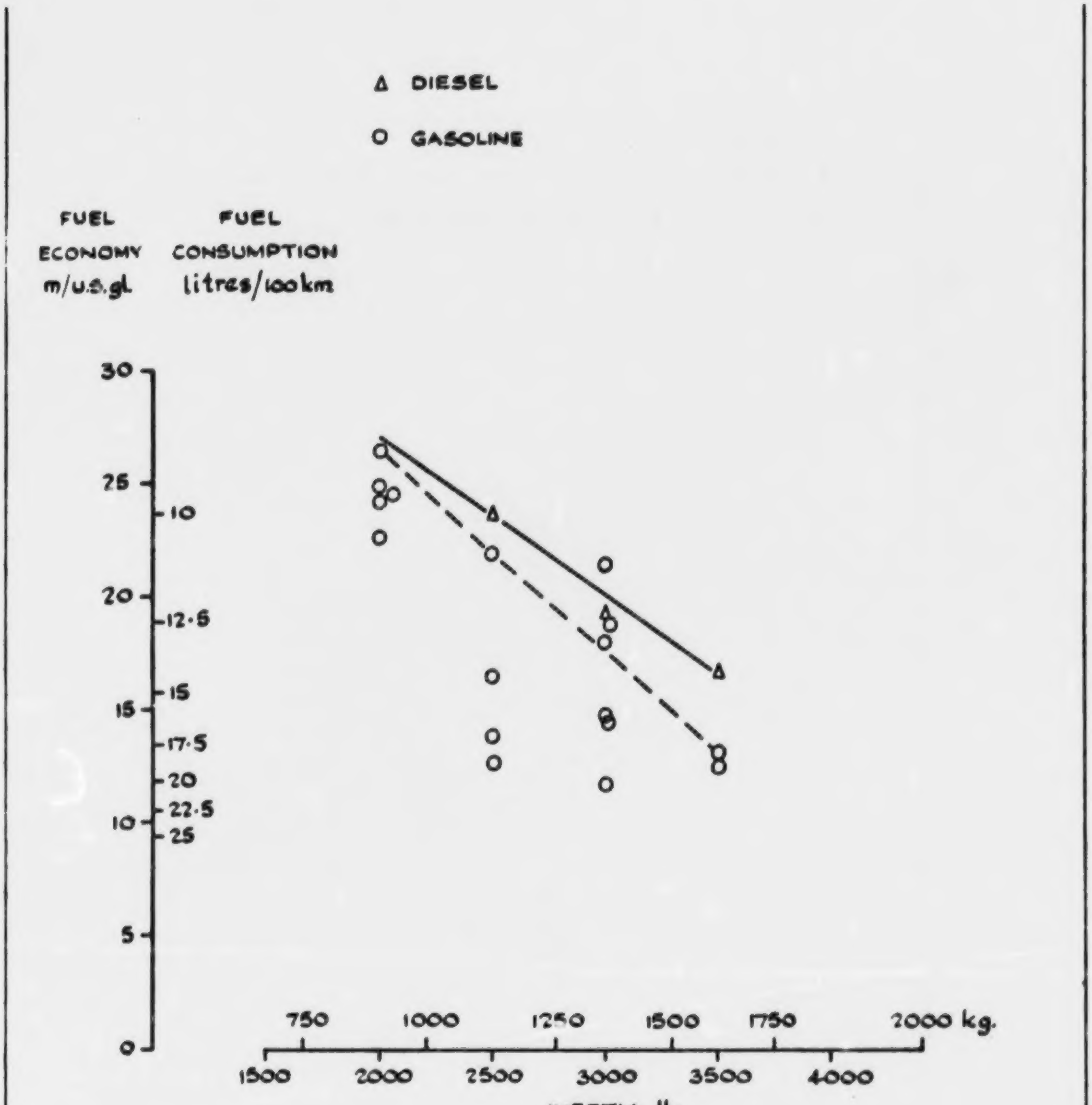
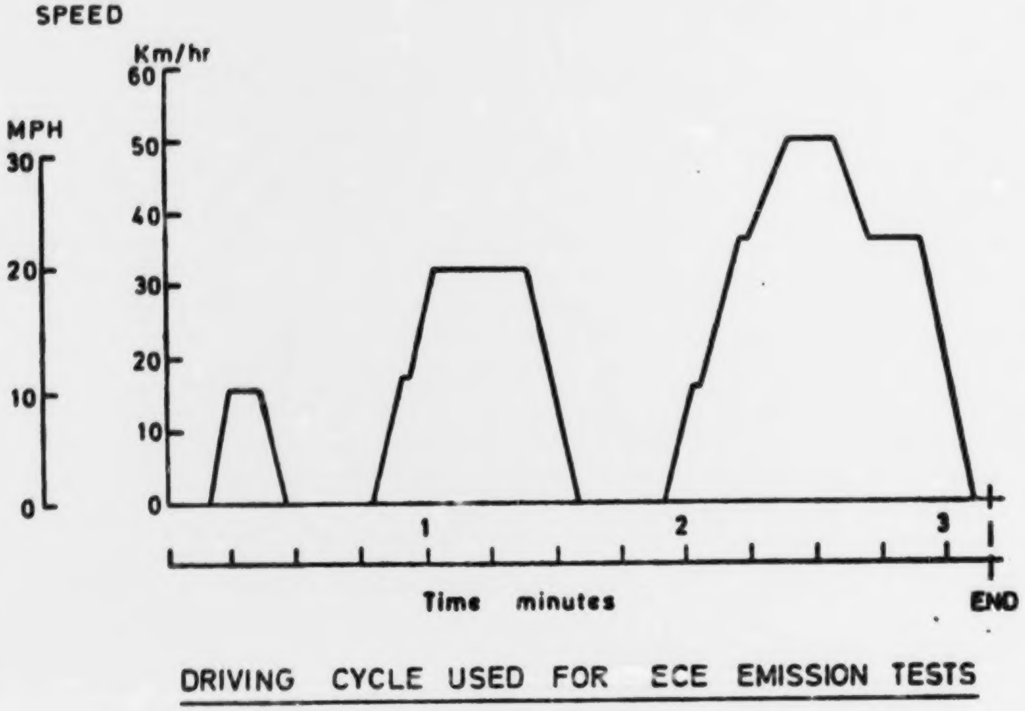


Figure A22. Driving cycle used for ECE emission tests.



Illus. 1643/9

Category 1

Single Combustion Chamber With Fuel Injection Before Combustion Mixture Formed By Fuel Injection And/Or Air Movement

History

The six principal types of combustion chambers reviewed in this section are shown in Fig 1.1 and 1.2. Of these, the Ford and MCP processes have been developed to the vehicle prototype stage, although the MCP was applied to agricultural equipment rather than automobiles. The Witsky system has been studied on test bed engines, and steady state exhaust emissions have been recorded. The fore-runner of these three engines was the Hesselman system, but only limited information is available, as the work was reported in the 1930's. Similarly, experience on the Starr engine, published in 1947 does not record exhaust emission data. The Windsor-Smith system is a relatively recent proposal, which has not been studied experimentally in any depth.

Topic Area (a) - Combustion Chamber and Stratification Technique

The Ford system, ref 1.1, depends mainly on the characteristics of the fuel injection to achieve stratification within a swirling air stream. A 90-115° cone of atomised fuel, injected near the centre of the swirl evaporates, and diffuses radially outwards. The air movement consists of swirl and also some squish generated by the piston cup. The period between the fuel injection and ignition in the Ford engine is controlled, such that the initial rich unit of fuel in the centre of the chamber has evaporated and diffused to a sufficient extent to be ignitable. Since ignition timing must be fixed to the region near tdc for good efficiency, timing of the fuel injection during the compression stroke must be varied, depending on the quantity of fuel injected and the engine speed. The Ford system has been developed along two separate lines, one an unthrottled version for good fuel economy, called FCP, and the other biased towards low emissions, and called Proco. In the former version, a long electrode spark plug located the ignition spark in the centre of the spray, i.e. in the region screened from direct fuel spray, by the head of the poppet valve of the injector. On the other hand, the Proco engine is throttled, so this region may be too rich to spark ignite and short spark plug electrodes can be used which do not protrude into the main injection stream.

The Proco engine operates at an air fuel ratio around 15 - 18 over the complete load range.

Both Ford systems depend heavily on the fuel injection equipment for achieving good stratification, while the other combustion systems in this category depend more on the air swirl movement for distributing the fuel. Nevertheless, in the others, the fuel injection characteristics are still important. In the Witsky, MCP and Hesselman engines, the fuel is injected either at the periphery of the air swirl, i.e., from the cylinder liner, or at some position close to the periphery, with the injector protruding from the cylinder head. The fuel droplets, fired through the air swirl, evaporate as they go, leaving a vapour wake which then rotates with the air swirl. Eventually, the droplets become so small that air viscous forces begin to predominate and the path of the droplet is controlled by the relative velocity of the droplet and the air.

Ref. G10 (Hussman et al) studied this complex evaporation process with the aid of a computer model, and the paper contains some interesting maps of the resulting mixture profiles within the chamber.

The mixture profile depends on the method and location of fuel injection. The three combustion systems being considered (i.e., Witsky, MCP and Hesselman Ref. 1,2 1,3 1,4 chose their respective injection and ignition locations on a trial and error basis. Witsky postulated that if the direction of fuel injection was against the air stream, then evaporating fuel droplets would eventually be drawn to the centre of the chamber, to form an ignitable mixture in this region. Therefore, the spark plug was placed in the centre of the chamber. Witsky experimented with injectors in the cylinder liner and in the cylinder head. During experiments with the MCP process, it was concluded that the centre of the chamber was not an ideal position and the spark plug should be offset. This conclusion was verified independently on theoretical grounds in Ref G10. The commonality between these engines was that injection was from a single hole nozzle, directed against the air swirl. The Hesselman engine used a two-spray injector, one spray against, and one following the air swirl. The spark plug and injector were both mounted in the cylinder liner.

The Starr engine(Ref. 1.5) represents a compromise between single and divided chambers. The chamber throat was large to prevent vigorous air movement and harsh combustion. The principal behind this approach was that the exhaust residuals remaining in the prechamber after the induction stroke, would be compressed into the region of the injector during compression. This hot gas would aid fuel evaporation, which could then be easily ignited by an adjacent spark plug. In addition, the squish action of the piston at tdc was thought to aid mixing once combustion had commenced.

Fuel Injection

A major advantage of this category of stratified charge engine is that fuel injection is virtually complete before combustion commences, and therefore the fuel injection pressure need only exceed gas compression pressure rather than combustion pressure. In theory the construction of a low pressure injection system should be less complex, resulting in a cheaper unit than its diesel equivalent. Indeed, Ford concluded that injection pressures in the region of 27.5 bar resulted in better atomisation and stratification, and this was substantiated in Ref G10. However, injection pumps with operating pressures in this region are not generally available, and the other experimenters in this area have obviously adopted high pressure diesel jerk pumps due to availability. Since Ford developed their own stratified charge pump and injectors, they were not constrained by availability. However, in spite of low pressures this injection system is not simple to manufacture. This stems from the sensitivity of the Ford combustion process to injection characteristics, such as timing, atomisation, etc. The injector is of the outward opening poppet valve type, and must oscillate on its seat at 2 - 4 kHz during injection. This aids in the break-up of the fuel stream. The moving component parts of the injector must be light to achieve this high natural frequency. On durability tests, carbon build-up at the nozzle was a problem, since the atomisation of the fuel was disturbed. This could only be relieved by placing close tolerances on the seating position of the injector poppet nozzle. Another critical area of the Ford process was injection timing, or to be more precise, the timing difference between injection and ignition. If this parameter was set incorrectly or wandered during service, exhaust emissions, driveability and fuel consumption could all suffer. Ford tackled this problem by developing an integrated distributor and injection jerk pump which has apparently been successful.

Witsky, MCP and Hesselman did not meet these problems, partly because they did not reach the stage of vehicle fleet application, and partly because diesel injection systems were adopted. High injection pressure apparently prevents carbon build-up on the critical injector components. If the potential for low injection pressures had been exploited in each of these engines, then similar problems might have been experienced. Similarly, the critical nature of phasing injection and ignition is common to all engines in this category, although the Ford system appears to be more sensitive than most. Nozzle types have been tailored to each engine. Atomisation of the fuel for quick evaporation and ignition is obviously a prime target. Witsky followed the same lines as Ford and developed a pulsed injector, giving two or three separate injections at low load and up to 12 at high load. MCP concentrated on a compound spray which gave a finely atomised wide spray at low loads and a heavier narrow spray at high loads. This guaranteed an ignitable mixture at the plug at all engine conditions.

Ignition

The difficulty in igniting stratified charge engines under all operating conditions has always been the main stumbling block during development. Unless the mixture is within an air fuel ratio range of 10 - 20 at the spark plug on every engine cycle, then the engine will mis-fire. Inevitably in a heterogeneous combustion chamber marginal conditions of ignitability will occur in some regions of the speed-load spectrum of the engine. To be sure of igniting under these conditions, a high energy and long duration spark is obligatory. Some researchers have observed a considerable extension of the engine operating spectrum when conventional ignition systems are replaced by high energy types.

Ford used conventional coil ignition, while Witsky chose a high voltage coil system. The MCP engine relied on a high voltage magneto. The references do not contain details on these units, but in-house experience with similar equipment suggests the following values:

	Voltage	Duration
Witsky, high voltage	30kV	1.0 m sec
MCP, magneto	25kV	0.7 m sec
Conventional coil	25kV	1.2 m sec

Air Swirl

Air swirl is a critical parameter in the stratification process which all the experimenters tailored to be a constant factor of the engine rotation speed. The combustion speed, in terms of engine crank angle, is often independent of engine speed. The following air swirl speeds were quoted in the reference:

Ford Proco	3-5 x engine rev/min
Witsky	2.5 x engine rev/min
MCP	2,0 x engine rev/min
Hesselman	not specified

These values should not be considered definitive since air swirl is measured by a variety of techniques which lead to different results. In particular, the swirl rig pressure drop adopted by Ford is considerably higher than is usual for diesel practice (Ford 0.116 bar, compared to 0.025 bar) which chould lead to higher calculated swirl levels. Generally the swirl in these engines was created with helical or directed inlet ports but this can be supplemented in other ways. Ford arranged for a portion of the inlet valve to be shrouded by the cylinder head at low lifts. Witsky incorporated a movable vane in the cylinder head, although this was only required at low engine speeds. Engine swirl level is a major parameter affecting the engine torque curve.

The four engines mentioned above all had piston bowls which accelerated the swirl rate during the compression stroke. The smaller the bowl the higher the final swirl rate. The bowls on the Witsky, MCP and Hesselman engines apparently occupying at least 50% of the bore area. In Ford engines the recesses are most compact occupying 35 - 42% of the bore area. There is also apparently some art in the exact design of the piston bowl, since the bowls can have a variety of lips, edges and corners. These details probably tailor the combustion chamber turbulence level to give the desired combustion speed. When injectors and spark plugs are inserted radially into a piston bowl combustion chamber (Witsky and Hesselman), there must be extra holes or slots in the piston crown.

Throttling

One of the motivating forces behind the efforts to produce a stratified charge engine has been the goal of low fuel consumption. This goal is compromised if the engine is throttled. However, generally some throttling to maintain approximately 30 : 1 air fuel ratio is permissible at light loads without seriously affecting fuel consumption. Stratified charge engines are often throttled and for a variety of reasons. The most common one is a "hole" in the engine spectrum, i.e., at some load or speed the engine either will not run or has a tendency to misfire. In other words, at this engine condition the fuel stratification is imperfect and the mixture at the spark plug is too lean to ignite. This can be cured by partial throttling of the engine to reduce the air flow and so increase the mixture strength at the plug. This factor is the principal difference between the original Ford stratified charge engine (called FCP) and the present throttled version called Proco.

Starr constructed a multi-cylinder engine based on the combustion system shown and found that it would not operate consistently below 25% load. Instead of throttling, an injection arrangement was provided to cut out some of the cylinders at low load. He claimed that the change-over to two cylinders was smooth, but it is doubtful whether this is a practical proposition. Ford throttled their second generation stratified charge engine Proco to an air fuel ratio of 15 - 18 : 1 at part load conditions in the cause of lower exhaust emissions. At full load the engine was enriched to 12.5 to 13.5 air fuel ratio. Throttling can be particularly beneficial in reducing hydrocarbon emissions. Hesselman experienced a light load smoke problem which was cured by throttling.

Throttling a stratified charge engine at idle can be beneficial for other reasons. The peak pressure and torque fluctuations are then reduced, thus alleviating vibration and noise. Since the consumption of fuel during idling is relatively small, poor thermal efficiencies are unimportant.

Compression Ratio

Stratified charge engines in this category can use higher compression ratios than the conventional engine, thus achieving a gain in indicated thermal efficiency. The combustion chamber end gas which contains air and fuel in the conventional engine is the source for spontaneous combustion. This source is removed in the stratified charge engine since the end gas contains mainly air.

The following compression ratios were used:

Ford	9.3 and 11
Witsky	9.5 and 12
MCP	8.0
Hesselman (1930)	5.5

At full load, the injected fuel must be mixed with as much air as possible to give a high torque without smoke emissions. Adequate mixing will obviously become more difficult as the engine bore size increases. Therefore, small bore/stroke ratios seem to be more suitable for fuel injected stratified charge engines. Swept volume also appears to have some influence. Ford experienced difficulty in applying their combustion process to a 8.75 l V8 engine and the compression ratio was reduced to 9:1. In addition, hydrocarbon emissions were high and the engine was prone to misfire. The development of the MCP process also suggests compatibility between small engines and stratified charge of this category since the engineers did not seem to have experienced great difficulty in applying the combustion system to 510 cc engines.

Topic Area (b) - Combustion Characteristics and Heat Rejection

The combustion of this type of stratified charge engine can be divided into two distinct phases, a fuel preparation stage and a burning stage. The fuel preparation stage is governed by fuel evaporation. This is perhaps the most important phase, and has led to many different theories to account for the behaviour of stratified charge engines. All the experimental evidence suggests that gasoline cannot be ignited and burnt in droplet form, and thus a droplet of fuel cannot be ignited by a spark before it has evaporated. Neither can the spark energy cause this evaporation because the resulting microscopic vapour charge could not contain air for combustion. Therefore, during the fuel preparation stage in a stratified charge engine the fuel must evaporate and mix with the air. This process requires time and, therefore, there must be a delay between injection and ignition. Furthermore, there will be a second delay, called ignition delay, while the spark forms a kernel of flame before the true combustion starts. These two delays, called evaporation and ignition delays are common to all combustion systems in this category. Typical delay values are given by Witsky (Ref. 1.2.1) at one engine condition: injection starts at 36° btdc and finishes after 11° of crank rotation; this is followed by 13° crank of fuel evaporation and then the spark occurs; ignition delay occupies a further 7° and then combustion commences. The evaporation delay period can be masked by long injection periods, so that combustion starts before injection finishes.

Turbulence and air movement in the combustion chamber rather than the type of injection, controls the flame speed. The speed of combustion has an influence on CO emissions and engine noise, and the development of Ford Proco has involved trade-offs in this area. In general, if there is insufficient air movement during and after combustion, areas of rich burning mixture will not fully oxidise to CO_2. Therefore, CO emissions can be high even in the presence of excess oxygen. If the turbulence level is increased CO is reduced and the cylinder pressure diagram takes the form of that shown in Fig. 1.3a. This diagram corresponds with injection timing for minimum fuel consumption. It might be expected that optimum efficiency for fast burning engines would correspond with combustion starting at tdc, since this approaches the ideal constant volume otto cycle. However, this is not the case. For engines with compression ratios in the range of interest, optimum efficiency is determined by the point of combustion finishing, which should be in the region of 20 - 30° atdc. The piston has only moved 6% of its stroke at 25° atdc so that the net work lost is small. If combustion finishes before 20° atdc the hot burning gas loses heat to the cylinder walls while waiting for the piston to commence the expansion stroke. Therefore, for engines with high turbulence and fast combustion the cylinder pressure may have passed the peak motoring value before combustion commences. This form of cylinder pressure diagram is undesirable with respect to combustion generated engine noise. Fig.1.3b.

The Witsky stratified charge engine was adapted for combustion photography through the piston. It was observed that the gas at the circumference of the combustion chamber did not take part in combustion even at full load. This was verified in the MCP engine by gas sampling which showed relatively weak unburnt mixtures in the end gas. This can lead to two problems:

1) high hydrocarbon emissions, and

2) a deficiency in engine power since all the air in the cylinder is not utilized.

These problems can be alleviated to some extent at full load by advancing the injection timing so that fuel evaporation is complete and a homogeneous mixture is formed in the combustion chamber, in a similar manner to conventional carburetted engines. At part loads very weak unburnt mixtures will remain at the edge of the combustion chamber and avoid oxidation during combustion. This will be expelled in the exhaust stroke as hydrocarbon emissions. As we shall see, this is the basic limitation of the stratified charge engines in this category.

At part load conditions in-cylinder gas temperatures are lower in unthrottled stratified charge engines than in the conventional gasoline engine and thus the heat losses to the coolant are also reduced under these conditions. Ford discovered that warm up of FCP engines was slow, and the coolant was arranged to flow only in the cylinder head during this period. An added incentive for fast warm up on this engine was the dependence of fuel consumption on coolant temperature.

Topic Area (c) - Performance

A favourable characteristic of this category of stratified charge engines is the relatively high torque if the injection timing is advanced at full load. A torque curve for the Proco engine scaled from test bed results on the four cylinder L141 engine is shown in Fig. 1.4. Good air utilization and low mechanical friction give a BMEP figure of 8.3 bar. Thus, the engine size to give the target horsepower is smaller than the diesel. The maximum engine speed was taken to be 4000 rpm as for the L141, although this could probably be improved with development, resulting in a further reduction in engine size. The falling torque as speed increases, is due to poor volumetric efficiency at higher engine speeds. This torque sacrifice is the payment for the high swirl requirements of the engine. The published literature on the Witsky system is rather vague on full load performance, although a figure of 10.1 bar IMEP (probably 8.6 bar BMEP in a multi-cylinder engine) is mentioned in the text. The performance of the MCP engine was inferior to both of these (Fig.1.5), giving a maximum BMEP of 7.1 bar. The reference does not mention any variation in injection timing with load, so it is presumed that this figure refers to constant injection timing. The performance could probably be improved by some form of injection timing modulation.

Engine friction is an important factor governing full load performance and low load fuel consumption. Since engine friction generally rises with compression ratio, the stratified charge engine with a compression ratio of 11 could have a friction level almost as low as a conventional gasoline engine, and considerably lower than a diesel. This is verified by figures from an early multi-cylinder FCP engine, see Fig. 1.6.

Vehicle performance of the Ford Proco version of the L141 engine was very poor, due to high EGR rates. This engine was tuned to meet the secondary emission targets and 10% EGR was used at full load. This reduced the specific engine performance to around 5.8 bar BMEP. In this build the engine could not follow the driving trace of the CVS-CH test, Fig. 1.7.

The CVS-CH test was simulated on a computer, using the parameters of the target vehicle, to find the minimum engine swept volume to follow the trace. It was shown that a Proco engine with a swept volume of 4.14 l in a 1600 kg car, could drive the cycle, and meet the primary emission targets. However, to achieve the secondary emission targets, the engine swept volume had to be increased to 5.25 l. This ensured that the engine power remained below 5.8 bar BMEP during the CVS-CH cycle. With this engine size there would be no restrictions on full load performance on the road, since the EGR would be modulated by a valve to zero flow. Therefore, the road performance of the target vehicle could be considerably better than that stipulated.

Customer acceptance tests on the MCP engine in an unspecified application (probably small agricultural) were favourable. The response of the engine to load control variations was judged to be good.

Topic Area (d) - Fuel Consumption

There is an impressive weight of evidence to justify the claims for better fuel economy from this class of stratified charge engine. When the unthrottled Ford FCP engine was installed in a vehicle normally powered with a gasoline engine, fuel consumption was reduced by 30% at steady state speeds of 30-70 miles per hour. Experience with the MCP engine suggests 20% improvement. In addition, members of the general public who had experience with the MCP claim fuel consumption reductions between 33 and 50%.

The results of fuel economy measurements during the CVS-CH tests for various Ford stratified charge engines are shown in Figs. 1.7 and 1.11. All the results in Fig. 1.7 refer to the L141 engine in the M151 utility truct (from reference 1.1.2). Car (b) corresponds to a best economy build of the Ford stratified charge engine (probably FCP), and the economy is 8% better than the trend line for good gasoline engines.

Results in Figs. 1.9, (A) of Fig. 1.10, and a Mk.IV Proco of Fig. 1.11 show fuel economy measurements from Proco vehicles without EGR, at NO_x levels close to the primary emissions targets. The levels are at least comparable with the good gasoline engines, and a significant improvement in economy was demonstrated in the larger vehicle. In addition, the 0.4 g/mile NO_x level was achieved with only a small loss in economy, Figs. 1.7, 1.8 and 1.10. The good economy at this level may arise due to the fast combustion properties of the Proco engine. In most other processes, the combustion speed is considerably reduced by EGR, resulting in a loss of cycle efficiency. However, the high turbulence level within the Proco engine, created by the air swirl, reduces this effect.

The test bed data on the Ford stratified charge engines leads to slightly different conclusions. Fig. 1.12 is a re-plot of the FCP results on a single cylinder engine, corrected to a friction level for a multi-cylinder FCP engine, and scaled to the target engine size. While the vehicle CVS results on the FCP give a fuel consumption midway between gasoline and diesel engines, the test bed information suggests FCP is better than the diesel. With throttling and EGR, the Proco engine has a fuel consumption 10% worse than the diesel (taken from test bed L141 results - Fig. 1.13).

Scaled brake fuel consumption figures of the MCP single cylinder engine are shown in Fig. 1.14. These could not be corrected to a multi-cylinder friction level, since the friction of the single cylinder engine was unknown. The higher friction horsepower of the single cylinder engine would cause a portion of this high fuel consumption. However, the form of the emission graph suggests that this stratification process is imperfect and this would contribute to the poor fuel consumption. Fuel consumption figures taken from a 4.28 l four cylinder Hesselman engine were very low compared to the diesel baseline, but the available data only covered one engine speed, Fig. 1.15.

Topic Area (e) - Exhaust Emissions

Unthrottled

Considering first the unthrottled stratified charge engines, FCP and MCP, scaled test bed exhaust emission data is shown in Figs. 1.12 and 1.14. In both cases the results were taken from single cylinder engines, although the FCP figures are corrected to a typical multi-cylinder engine friction. The single cylinder FCP engine had a swept volume of 880 cc, while the MCP engine was 570 cc. Although the general HC and NO characteristics of the two engines were similar, the level of HC emissions was greater with the MCP. Data from the Witsky engine was not reported in the same format but the results show the same excessive hydrocarbon level. In-cylinder tests on the MCP engine with a timed sampling valve, showed that the exhaust HC emissions could be related to the mixture strength of the charge in the end gas, i.e. at the furthermost point from the spark plug. In other words, there is a portion of the cylinder charge at the boundary of the combustion chamber, which is not consumed by the combustion flame because it is too weak. Nitrogen Oxide emissions are universally low from unthrottled engines.

It might be presumed that the high exhaust hydrocarbons could be oxidised in a catalytic muffler, but this is not completely true. At low loads the exhaust temperature of the unthrottled stratified charge engine is in the region of 200 - 300°C compared with 600 - 800°C for an engine throttled to stoichiometry. Since exhaust catalytic reactors generally only begin to function at 300°C, there will be a considerable portion of the operating range of the stratified charge engine over which the exhaust emissions cannot be controlled. In addition, the time to "light-off" in the CVS-CH test will be relatively long, and the cold start HC emissions will be high. This problem of catalyst application is common to all unthrottled engines.

Many researchers in this field have observed a characteristic odour from stratified charge engines which is thought to be formaldehyde or benzaldehyde. These compounds form from the partial oxidation of gasoline. Vanadium pentoxide catalysts have been used by Ford to control this phenomenon with some success. Ricardo experience with diesels suggests that platinum based catalysts are effective in reducing odour at low mileage. However, in long term durability tests on gasoline engines these same catalysts can generate a benzaldehyde odour from the normally odour-less hydrocarbon emission of the engine. These three problem areas of hydrocarbon emissions, catalyst inactivity and odour associated with this category of unthrottled stratified charge engines can be cured by partial throttling.

Throttled

The Ford Proco engine is the only developed example of a throttled stratified charge engine in this category, and most of the following comments refer to published results and experience of this engine. The engine operates at an air/fuel ratio in the region of 15 - 18 : 1 throughout the engine load range. This engine has been developed to meet low NO_x targets which are beyond the capability of the conventional gasoline engine. This is possible due to the tolerance of the

Proco engine to high EGR levels up to 25% of the air flow. Exhaust gas recirculation rates of this magnitude always involve some increase in hydrocarbon emissions. Therefore, the application of throttling and EGR to the Ford combustion process can show HC reduction with throttling which is then compromised by the EGR. Emission figures from the L141 Proco engine with 15% EGR are shown in Fig. 1.13.

Hydrocarbon levels are worse than many conventional gasoline engines but nitric oxide emissions are exceptionally low. The trade-off between throttling and EGR level is very critical. A Proco vehicle was tuned for two different NO_x levels in the CVS-CH test, i.e., 0.35 g/mile and 1.1 g/mile. This increase in NO_xmass emission yields an HC emission reduction from 3.5 to 1.3 g/mile. CO emissions from the Ford Proco can be a problem, due to areas of rich mixture within the chamber. Inspite of overall lean mixtures CO concentration levels can vary between 0.15 and 1.5% compared with less than 0,1% from a conventional gasoline engine.

The primary emission target has not been achieved by any PROCO vehicle without a catalyst reactor, although one M151 engine came relatively close on all 3 emission components (Fig. 1.10 (A)). When a catalyst was applied, the target could be achieved with ease (Fig. 1.8 (D)).

Using catalysts and EGR, NO_x levels within the secondary emission targets could be consistently achieved, and 0.24 g/mile has been recorded. In addition, reference 16 considers that the standard deviation in emissions from fleets of stratified charge engines is less than that of conventional engines. It is apparent from car (A) in Fig. 1.7 and (B) and (C) of Fig. 1.10. that baseline HC emissions are high at low NO_x levels and that considerable HC control is required from the catalytic reactor. Therefore, the Proco engine relies on the durability performance of a catalytic reactor to a greater extent than in the conventional gasoline engine. During a 50,000 mile durability test two changes of reactor were necessary to keep the engine within the secondary emission targets. The main advantage of the Proco engine over the gasoline engine in this respect is the absence of an air pump. The exhaust gas from the engine contains sufficient oxygen to complete the oxidation of the hydrocarbons and carbon monoxide.

Other Emissions

Smoke records taken over a CVS-CH test show lower levels from a Proco Capri than most other types of stratified charge engines, Fig. 2.28. No observations were reported on smoke emissions of the Witsky or MCP engines but some light load smoke was emitted from the Hesselman engine. Tests conducted by Dow Corning on the Proco Capri indicated, as expected, that particulate emissions were only marginally greater than those of most gasoline engines. Odour levels from this car were also at such a low level, compared to a diesel engine, that difficulties arose during measurement. The odour measurement technique was judgement by a human odour panel, and the results were presented as composite odour intensity. Chemical analysis of exhaust gas revealed similar levels of aldehyde and ammonia emissions from the Proco engine, as from a conventional catalyst equipped gasoline engine.

Topic Area (f) - Multi Fuel Capability

The references do not mention any multi fuel capability for the Ford stratified charge engines and private communications have substantiated this.

These engines have been developed primarily for low emissions and the critical timings of injection and ignition, when set for gasoline may not be applicable to other fuels. However, the Ford engines are apparently the exception in this respect. The Witsky engine operated equally well on a combat gasoline, JP4, CITE fuel kerosene and diesel No. 2, and the fuels could be interchanged during running with no detectable power change of the engine. Hesselman also found that his engine tolerated non-volatile and heavy oil fuels. Successful cold starting tests were performed on the MCP engine using both gasoline and kerosene down to temperatures of - 30°C.

Topic Area (g) - Engine Components & Manufacturability

Since the Ford stratified engines are the only examples in this category which were tested extensively in vehicles, most of the comments below refer to this type. The achilles heel of the Proco and FCP engines is the injectors, since stratification and combustion both depend on the injectors working correctly. However, in durability tests coking and fouling by combustion deposits have been experienced. In addition, the manufacturing tolerances are tight since spray concentricity must be precise, and the quantity of fuel delivered must be repeatable on each engine cycle. In common with all fuel injection systems, the standard of fuel filtration must be high. These manufacturing difficulties have led Ford to conduct a large scale feasibility study on the injection equipment for stratified charge engines. Results have not yet been published.

The integrated pump/distributor unit used on the L141 appears to have been successful, although there was some wear on the face cam which drives the plungers. The unit and its controls with manifold vacuum, engine speed, EGR and driver command could be difficult to set up and maintain. The complexity of the unit will also influence engine cost.

The ignition system on the FCP caused plug wear rates 4 to 8 times greater than in carburetted engines. This could be alleviated to some extent with larger area electrodes and precious metal materials. The shorter electrodes in the Proco also help, but higher wear rates could still be expected on this engine.

During the rather limited durability tests of the Witsky and MCP engine the problems experienced by Ford were not observed. In both cases, the condition of the spark plug remained good.

Topic Area (i) - Mathematical Models

Bracco's Paper, Ref. G6 is a unique classification of computer models for reciprocating and rotary internal combustion engines, which is particularly orientated towards stratified charge engines. Most of the existing computer programmes are divided into two categories, zero dimensional and one-dimensional models. The zero dimensional model considers the engine cycle in a thermodynamic manner without air or fuel movement. The combustion is approximated as some function of the heat released by burning, against crank angle. On the other hand, the one-dimensional model will consider the inter-action between fuel and air drops due to their respective velocities during the injection process. In addition, the flame speed during combustion is estimated from the scale of turbulence and empirical models are not required. The author of the above reference states that the one-dimensional model is much more useful since combustion speeds and cycle efficiencies for new combustion chamber designs can be calculated without extrapolating empirical combustion approximations. This is not true for the conventional gasoline engine, since an experienced combustion engineer can probably guess combustion speed more accurately than the turbulence predictions from the engine configuration. Nevertheless, the more complex one-dimensional model is justifiable for stratified charge, since the mixture stratification within the cylinder cannot easily be guessed. Computer predictions can help in choosing the optimum spark plug position rather than experimenting with a large number of positions.

The computer stratification model in reference G10 was based on fuel droplet size and velocity, and air swirl. The droplet was acting under gas viscosity and droplet inertia forces and the mass change due to evaporation. The droplet trajectory was marked by a vapour wake. By considering a family of droplet sizes, the final stratification function within the cylinder was plotted. This model applies mainly to the Hesselman, MCP and Witsky engines.

One of the papers in reference G1 is devoted to a zero-dimensional computer model of the Ford Proco process. Empirical relationships are included for the stratification function within the cylinder and the cumulative mass burn. A cosine formula is assumed for cumulative mass burn against crank angle. The model effectively shows that the cylinder stratification should be destroyed after combustion, i.e., the gases should be mixed so that any rich regions with a CO content can react with regions containing oxygen. In addition, the stratification function should be arranged such that the spark plug ignites either a rich or a lean mixture. If the mixture at the plug is stoichiometric NO_x emissions increase. The author also concludes that high exhaust gas recirculation rates are more beneficial for low NO_x than lean mixtures. In other words, for low NO_x it is better to operate a stratified charge engine at an air fuel ratio of 18 and 15% EGR, than an air fuel ratio of 22.5 and no EGR.

The former option would involve a sacrifice in fuel consumption, although this may be small if the combustion speed were not affected.

Topic Area (j) - Engine Size and Weight

Existing examples of multi-cylinder stratified charge engines are modified versions of production gasoline engines. Therefore, there are no definitive figures of size and weight for specifically designed stratified charge engines. The four-cylinder Proco engines probably weighed about 12 kg more than the standard gasoline counterpart, due to the injection equipment and the larger pistons. The base engine would weigh about 150 kg. However, if this engine were designed specifically to operate on the Proco principle, there would be an additional weight penalty of about 1 kg due to a taller block and larger balance weights on the crankshaft.

Topic Area (k) - Noise and Vibration

Engine noise originates from two sources; combustion and that which is mechanically induced. For stratified charge engines in this category noise is a trade-off area with engine performance and fuel consumption. All of the engines considered require air movement for stratification of the fuel, but turbulence is inherently associated with air movement, and combustion speeds increase as turbulence increases. Therefore, rates of pressure rise in the cylinder which are the source for combustion induced noise, are often higher than desirable. 2.0 bar/° crank angle is generally considered to be a satisfactory rate of pressure rise for a gasoline engine. The following rates have been experienced in unthrottled stratified charge engines in various builds:

FCP	1.7 - 3.1 Bar/° crank angle
Witsky	0.8 - 2.9 Bar/° crank angle

Rates of pressure rise can be reduced by throttling but the fuel stratification then becomes richer, and higher CO emissions may result. To reduce the CO, cylinder turbulence levels must be increased, so that some of the throttling action has been lost. EGR also exerts an influence in slowing the combustion rate. Noise cannot be considered to be a simple function of rate of pressure rise because cycle to cycle variations in combustion rate influence the subjective combustion harshness. As rates of pressure rise increase, cycle to cycle peak pressure variations and subjective engine noise may, in fact decrease.

A detailed inspection of a Proco engine (by EPA) revealed large piston to cylinder clearance. This was presumably dictated by high piston thermal loadings at full load, due to combustion taking place in the piston bowl, and high heat transfer coefficients due to forced convection from the charge swirl. Therefore, piston slap may be a mechanical noise source of Proco engines, if conventional piston profiles are employed.

It may be seen from the foregoing explanation that noise from stratified charge engines and Proco in particular, is a complex phenomena.

SWRI, Ref. G9, have conducted tests on a number of vehicles, among which was a Proco Capri. The idle noise of the Proco was higher than a similar gasoline engine, although significantly lower than unthrottled engines such as the diesel. Information from Ford suggests that the idle noise level has been reduced in later engines. At steady vehicle speeds and low engine loads, the Proco noise level was not a problem, but at full load the noise was higher than any other engine studied. Full load noise is influenced by many factors, and the bore sizes and maximum speeds of the engines studied, varied considerably. Therefore, Ricardo used in-house prediction techniques to determine the probable noise level of a gasoline or diesel engine at the same bore size and speed as the test engine. Even by this technique, the full load noise of the Proco was higher than any gasoline or diesel engine, possibly due to either very fast combustion or piston slap.

If the primary source is piston slap then detailed heat transfer studies and further piston development could cure this problem. However, if the source lies in the combustion mechanism, then palliative techniques for engine stiffening, panel damping and engine isolation must be applied.

The MCP engine was less noisy than its carburetted counterpart and this may be linked with the lower air swirl rate (two times engine rpm compared with 3 - 5 times engine rpm for Proco).

Engine vibration and noise at idle are common complaints of unthrottled stratified charge engines. The maximum cylinder pressures are much greater than in throttled engines, leading to large torque fluctuations during an engine cycle. Since combustion in stratified charge engines can often be improved with throttling (unlike the diesel), there is a good case for universal throttling at idle.

Topic Area (1) - Cold Starting

In-cylinder fuel injection gives an impressive advantage to stratified charge with respect to starting and driveability. Enrichment during start-up is neither necessary nor desirable at low temperatures. The Proco engine could be started at - 23°C after 1 - 4 seconds cranking. When fuel was added to the induction system successful starts were achieved at - 29°C. In addition, under cold or warm conditions, load response of the engine was good since liquid fuel was not present in the manifold as in a carburetted engine. These characteristics have been observed from a number of the engines in this category.

High cranking speeds were required on Proco engines to overcome fuel leakage past the injection pump plungers and to produce the correct fuel spray characteristic.

Patents

The Windsor-Smith patent where the piston forms a prechamber at TDC is an interesting concept. This arrangement might alleviate the excessive hydrocarbon emissions of unthrottled engines at light loads. The fuel would be injected only into the prechamber and thus would burn homogeneously in this region. The lean peripheral area of the single chamber engine would not exist. In addition, by injecting fuel near bdc, complete air utilization could be achieved at full load. The difficult operating region would be in the mid-load range where these two regimes must be joined.

Fuel injected during the middle of the compression stroke would be distributed between the two chambers. The smaller chamber would burn, but the larger chamber would still be very lean overall, and would not support combustion of all the hydrocarbons. Therefore, in the mid-load range this engine might have the same hydrocarbon emissions as FCP or MCP.

An extra bonus of this sytem is the absence of air swirl which would not compromise the engine torque curve and maximum power output. In addition, if the throat size between the chamber was chosen correctly combustion noise might be lower than the other engines in this category, without the attendant trade-off in CO emissions.

Piston thermal loading might still be a problem area due to high gas velocities over the piston crown.

Turbocharging

Turbocharging is a popular method of increasing specific power output of large diesel engines. In theory turbochargers can also be applied to unthrottled stratified charge engines. However, as has been shown, the unthrottled engines considered are unacceptable on the grounds of high hydrocarbon emissions, and this situation would not be altered by using a turbocharger. When throttled stratified charge engines are boosted, the maximum engine power can be improved. However, this single advantage must be balanced against a number of disadvantages which apply equally to the turbocharging of throttled conventional gasoline engines.

1) Stratified charge engines using early fuel injection are sensitive to knock, although the CR is usually one or two ratios above the gasoline engine. Therefore, if these engines were turbocharged the compression ratio would be reduced to inhibit knock. Thus some of the efficiency gain of turbocharging would be lost.

2) Pressure boosting occures mainly in the medium and high speed range with little advantage in city driving.

3) At high speeds a waste gate is required to relieve the exhaust pressure before the turbine, or a boost control valve to relieve the inlet pressure after the compressor.

4) There is little gain in fuel consumption over the naturally aspirated engine unless the specific torque output of the naturally aspirated engine is relatively low (below 6.9 bar BMEP). The Ford Proco engine has a BMEP in the region of 8.2 bar.

5) NO_x emissions may be increased.

6) The response time of a turbocharger is finite, and driving patterns must be modified to accommodate this.

In view of the above factors, it is difficult to justify using a turbocharger except where space for installing the engine is limited and increase in engine size and swept volume cannot be tolerated.

Application to Motorcycles

The effect of applying the combustion systems in this category to the small cylinder sizes is unknown, but wetting of the combustion chamber wall could probably not be avoided during injection. In addition, the surface to volume ratio of the combustion chamber would be greater than in larger sizes, increasing the heat transfer. Therefore, utilization of the fuel would be inferior to that of larger engines, although still considerably better than that of gasoline engines operating at rich mixtures. Motorcycle engines typically produce 50 to 60 kW/litre at about 8500 to 10,000 rev/min, over a wide range of cylinder sizes. The maximum practical speed of present injection systems corresponds to an engine speed of 5000 rev/min. and thus if this type of stratified charge combustion were applied to a motorcycle a drop in specific output to below 28 kW/litre could not be avoided.

HC emissions from a PROCO combustion chamber of about 200 cc swept volume would certainly be higher than the present 500-900 cc versions. However, the engine would still meet the motorcycle target emission levels for HC, CO and NO_x without a catalytic device or EGR.

The cost of the medium pressure injection system for a motorcycle would easily exceed the cost of the bare engine and this would prejudice the application to small motorcycles. These combustion systems have not been applied to air cooled engines, although no potential problems are envisaged.

Conclusions from Category 1

1) Stratification within a direct injection engine can only be achieved with difficulty and requires air movement and an injection system tailored to that engine. The combustion systems are always sensitive to the correct settings of injection and ignition timing and if these settings stray far from optimum the engine misfires.

2) Unthrottled stratified charge engines will give a fuel economy better than a gasoline engine and approaching that of the diesel. However, all of the engines reviewed have a fundamental limitation of high hydrocarbon emission, which apparently cannot be avoided. In addition, oxidation catalysts at their present stage of development could not

remove a sufficient quantity of hydrocarbons for the target vehicle to reach the primary emission targets. If this fundamental limitation is to be avoided a completely new line of development must be followed. The Windsor-Smith patent is an example of a new approach but even here a successful outcome is far from certain.

3) If the stratified charge engines are throttled then the hydrocarbon emissions can be reduced and controlled, but at some sacrifice in fuel economy. The Ford Proco engine is the only example of this approach which has reached a stage of development where it can be considered as a viable automotive power plant.

4) By applying a catalyst and some EGR to a Proco engine, the primary emission target can be attained. The catalyst life is likely to be longer than on a conventional engine and the vehicle fuel economy would be as good or better than conventional engines with a NO_x level of 2 - 4 g/mile.

5) Using high levels of EGR the exhaust emissions of the Proco engine can be brought within the secondary emission targets at zero miles. There are only a limited number of types of internal combustion engines which can achieve 0.4 g/mile NO_x with known technology. At this NO_x level the bare engine hydrocarbon level is high and adequate catalyst life may be a problem, where high conversion efficiencies are required over an extended period. Vehicle fuel economy is impressive and does not differ appreciably from good gasoline engines without emission controls.

6) The other advantages of this category of stratified charge engine are as follows:

   (a) Good driveability and excellent cold starting.

   (b) With a suitable arrangement of injector and spark plug some multi-fuel capability can be demonstrated. This does not apply to Proco since it was specifically developed for gasoline. Operation on a variety of fuels will probably involve some sacrifice in exhaust emissions.

7) The disadvantages of this category of stratified charge engine are as follows:

   (a) Engine swept volume will be slightly greater than a conventional gasoline engine, but less than a diesel. A further increase in swept volume may be necessary to achieve the secondary emission targets.

   (b) The engine is noisier than the gasoline engine, particularly at idle. This area requires further study before any conclusions on the potential for noise reductions can be drawn.

   (c) The principal problem area for the Proco engine is mass production of the injectors. There is apparently some doubt as to whether the required tolerances can be held during production.

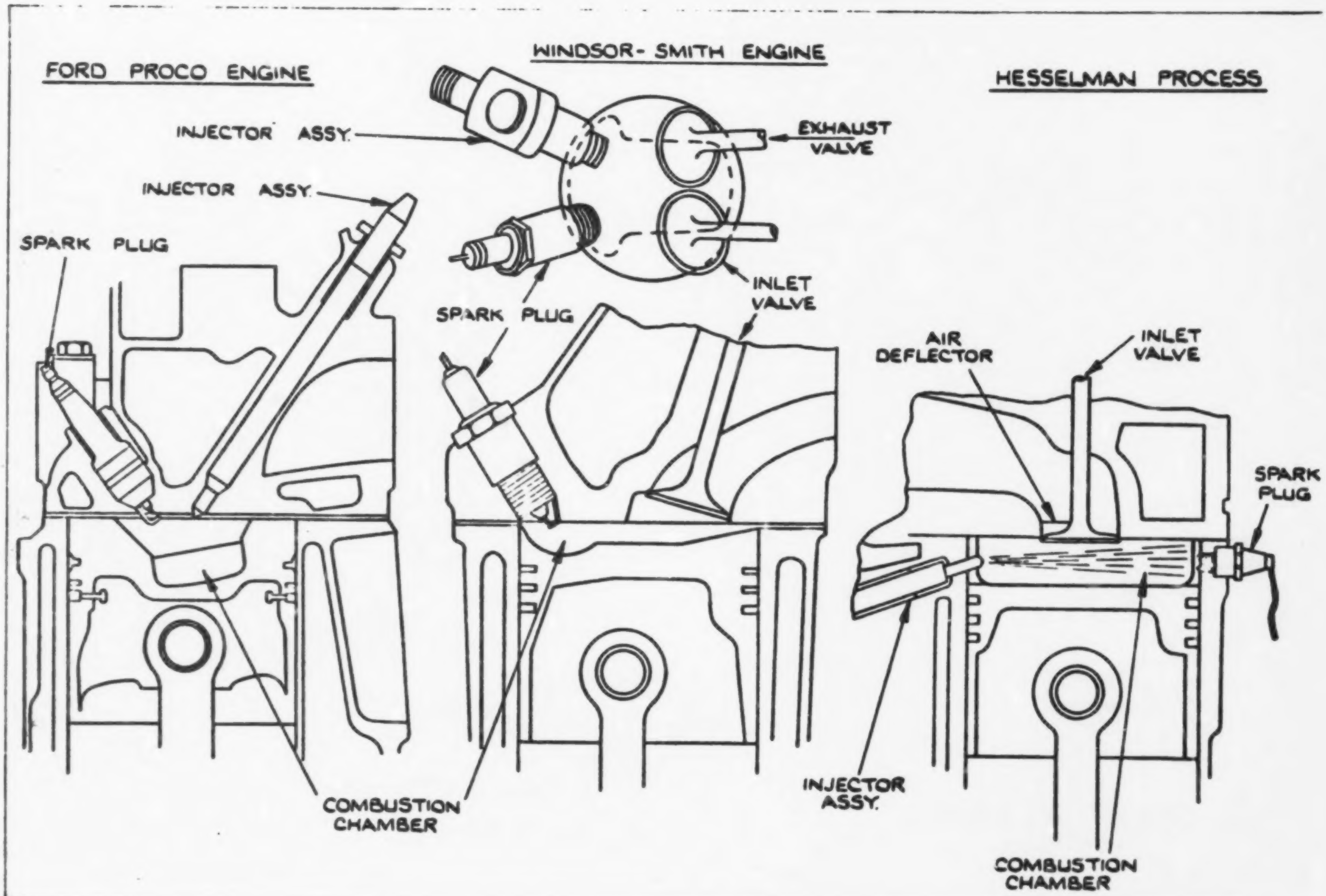

FORD PROCO ENGINE
INJECTOR ASSY.
INJECTOR ASSY.
SPARK PLUG
COMBUSTION CHAMBER
WINDSOR- SMITH ENGINE
EXHAUST VALVE
SPARK PLUG
INLET VALVE
SPARK PLUG
HESSELMAN PROCESS
AIR DEFLECTOR
INLET VALVE
SPARK PLUG
INJECTOR ASSY.
COMBUSTION CHAMBER

Figure 1.1 Proco, Smith and Hesselman engines.

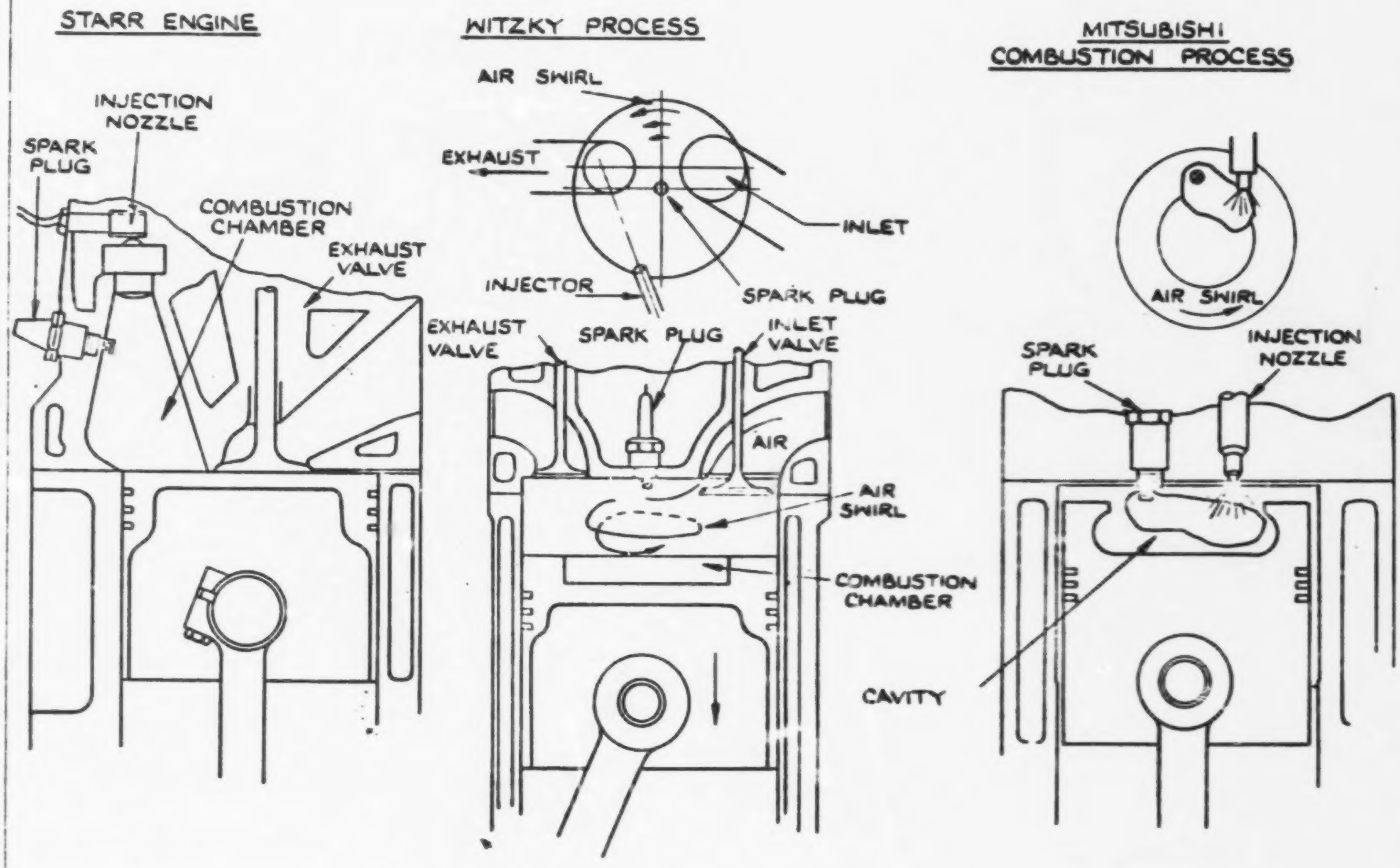
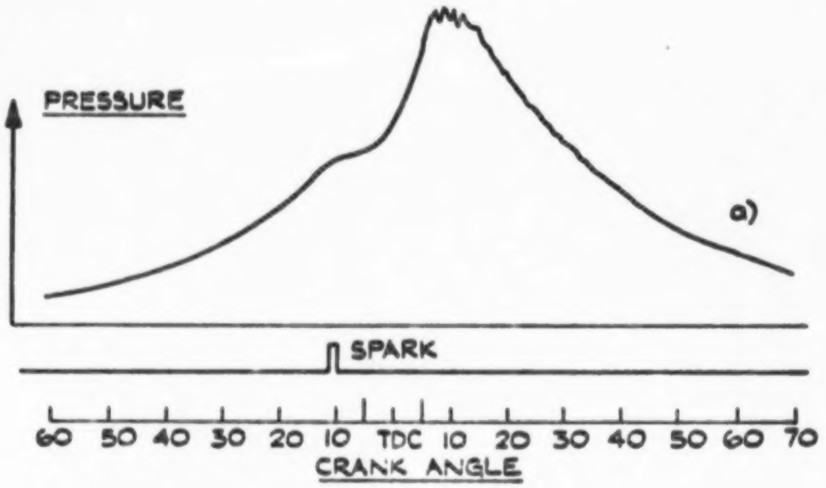

STARR ENGINE
INJECTION NOZZLE
SPARK PLUG
COMBUSTION CHAMBER
EXHAUST VALVE
WITZKY PROCESS
AIR SWIRL
EXHAUST
INLET
INJECTOR
SPARK PLUG
EXHAUST VALVE
SPARK PLUG
INLET VALVE
AIR
AIR SWIRL
COMBUSTION CHAMBER
MITSUBISHI COMBUSTION PROCESS
AIR SWIRL
SPARK PLUG
INJECTION NOZZLE
CAVITY

Figure 1.2 Starr, Witsky and Mitsubishi engines.

Figure 1.3. Cylinder pressure diagrams from a stratified charge engine.

6·5 BAR b.m.e.p. 2000 rev/min IGNITION AND
INJECTION TIMING FOR BEST TORQUE.



a) SLOW COMBUSTION

b) COMBUSTION CHAMBER PARAMETERS ALTERED TO INCREASE COMBUSTION SPEED.

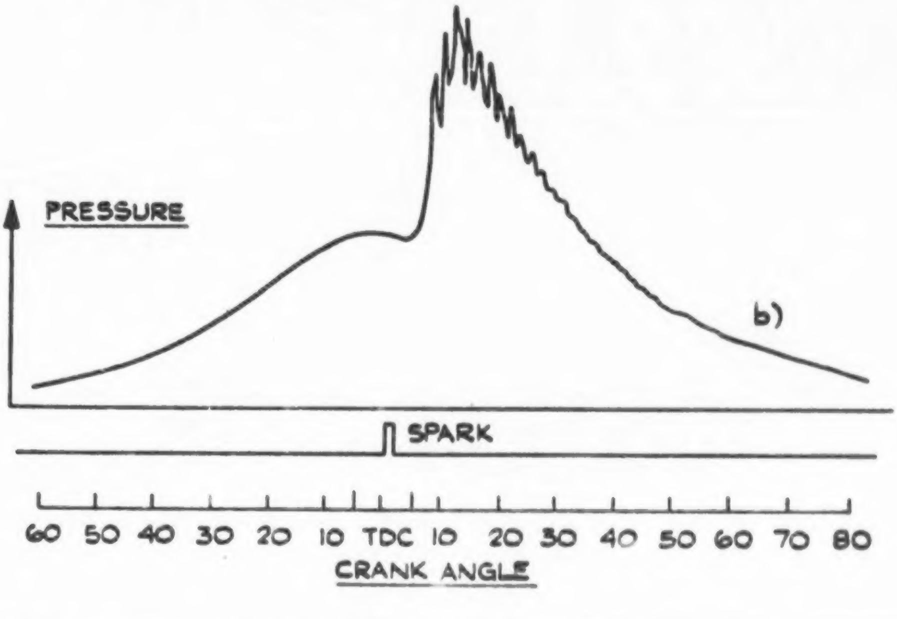


Figure 1.4. Comparison of torque curves Ford Proco and baseline diesel. (Proco figures extrapolated from L141.)

PROCO ENGINE
SWEPT VOL FOR 96 kW = 4·16l (254 in^3) MAX POWER SPEED = 4000 rev/min.
DIESEL ENGINE
SWEPT VOL FOR 96 kW = 4·77l (291 in^3) MAX POWER SPEED = 4000 rev/min.

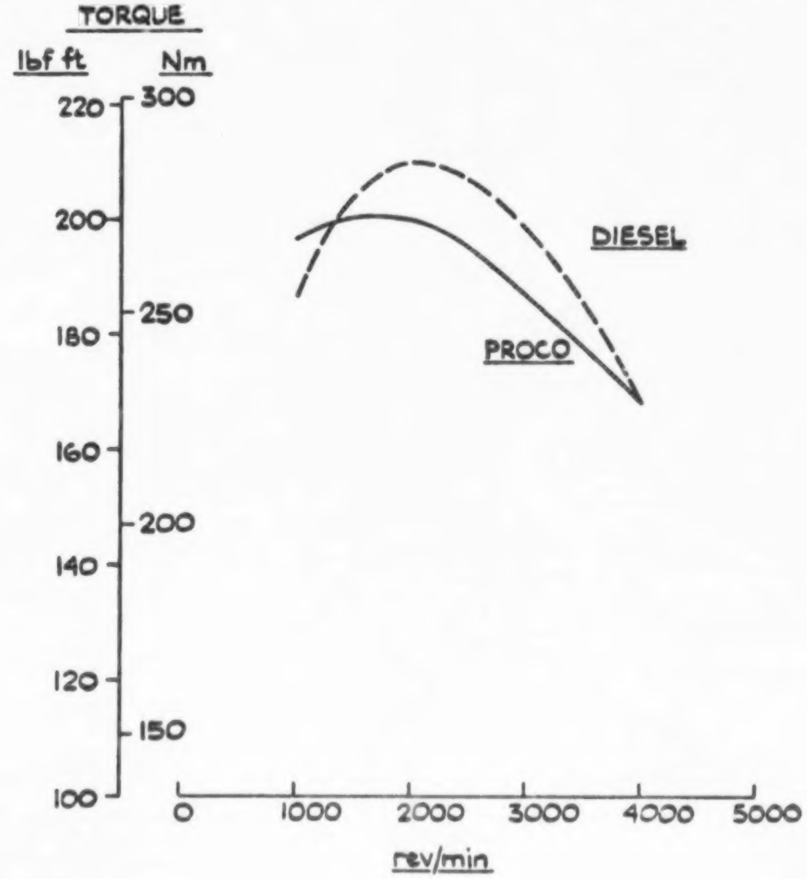


Figure 1.5. Comparison of torque curves of baseline diesel engine and MCP engine (data scaled from single cylinder and not corrected for friction).

DIESEL ENGINE
SWEPT VOL FOR 96kW = 4·77l (291 in³) MAX POWER SPEED = 4000 rev/min.

MCP ENGINE
SWEPT VOL FOR 96kW = 5·04l (307 in³) MAX POWER SPEED = 4000 rev/min.

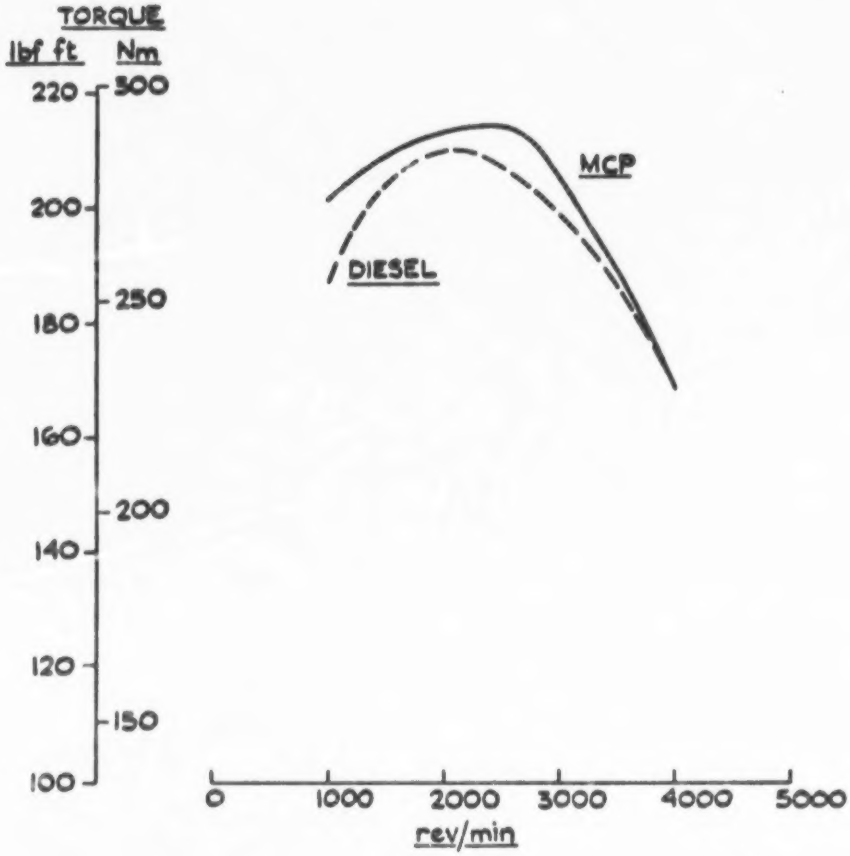


Figure 1.6. Motoring friction of FCP engine in comparison to diesel and petrol engines.

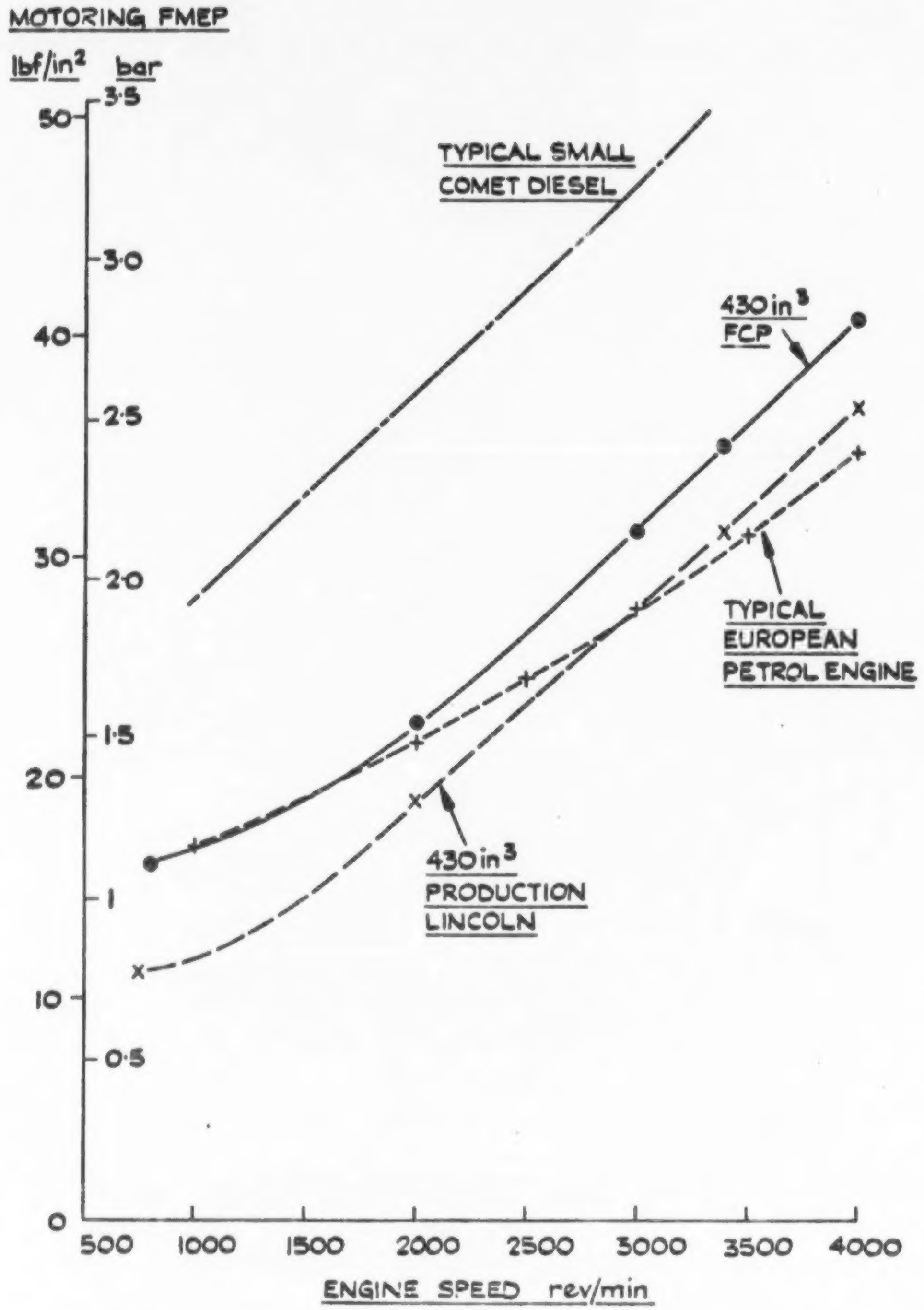


Figure 1.7. CVS-CH emission and fuel consumption results from Ford.

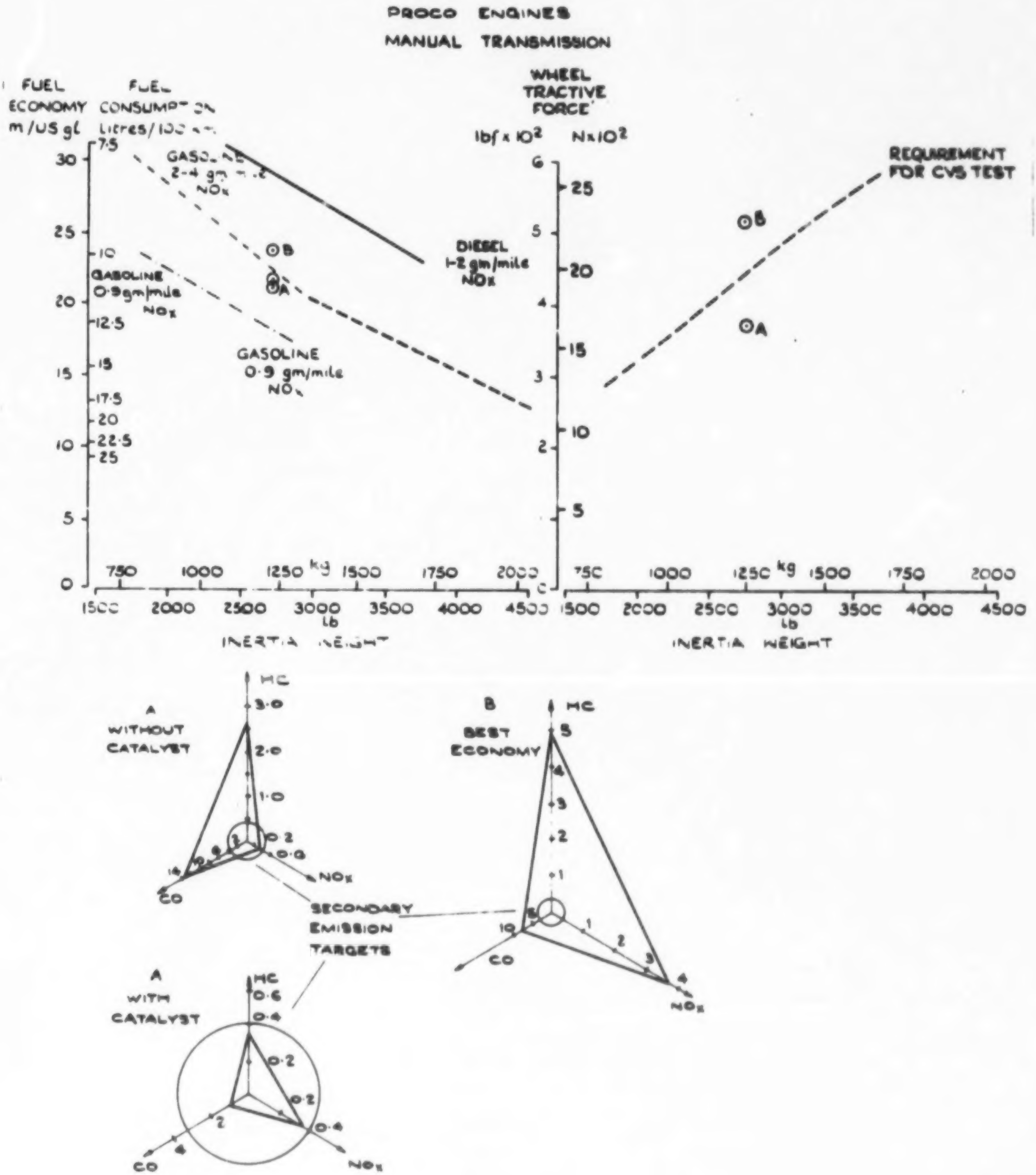


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Figure 1.8. Emission and fuel consumption results from Ford Proco engines.

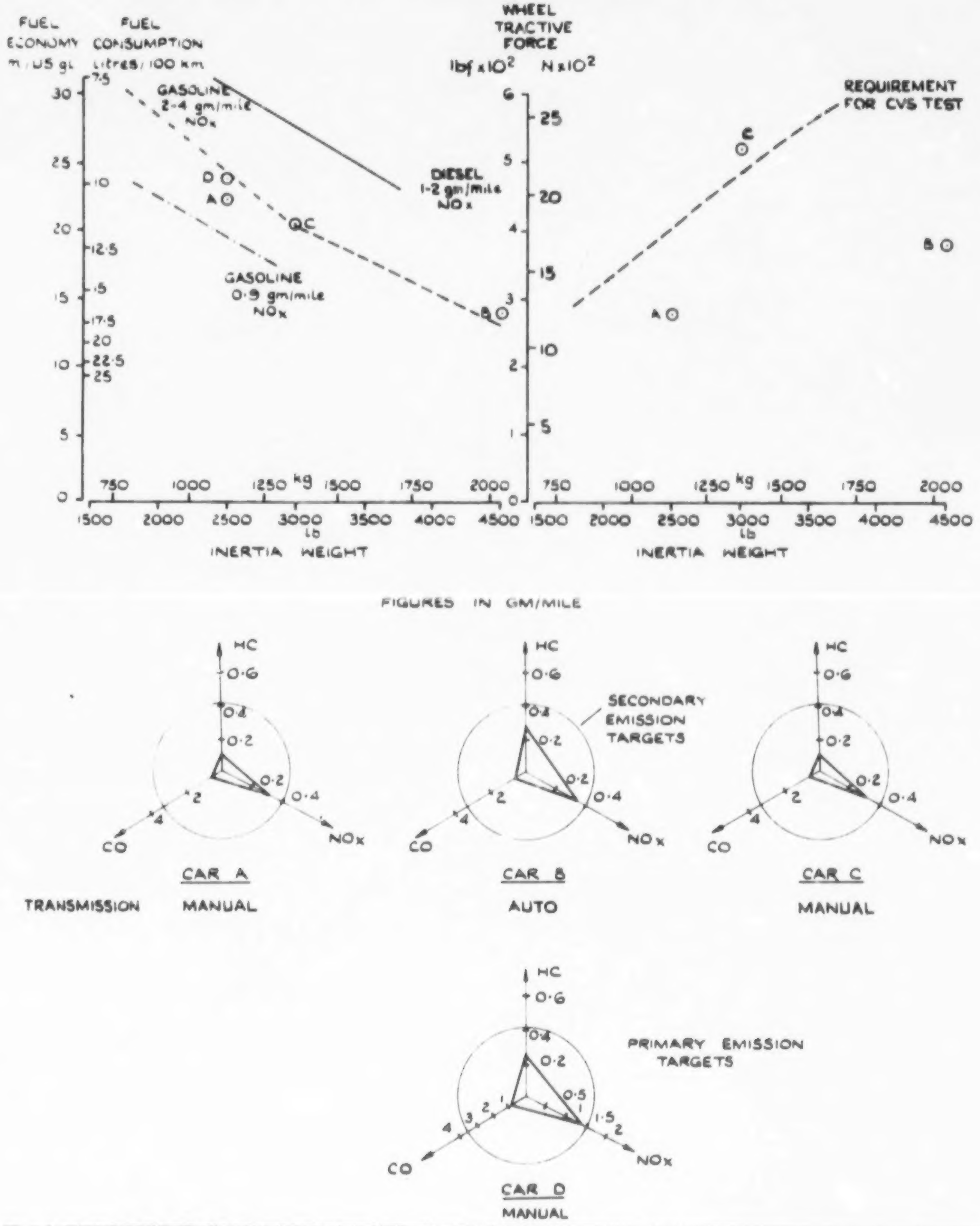
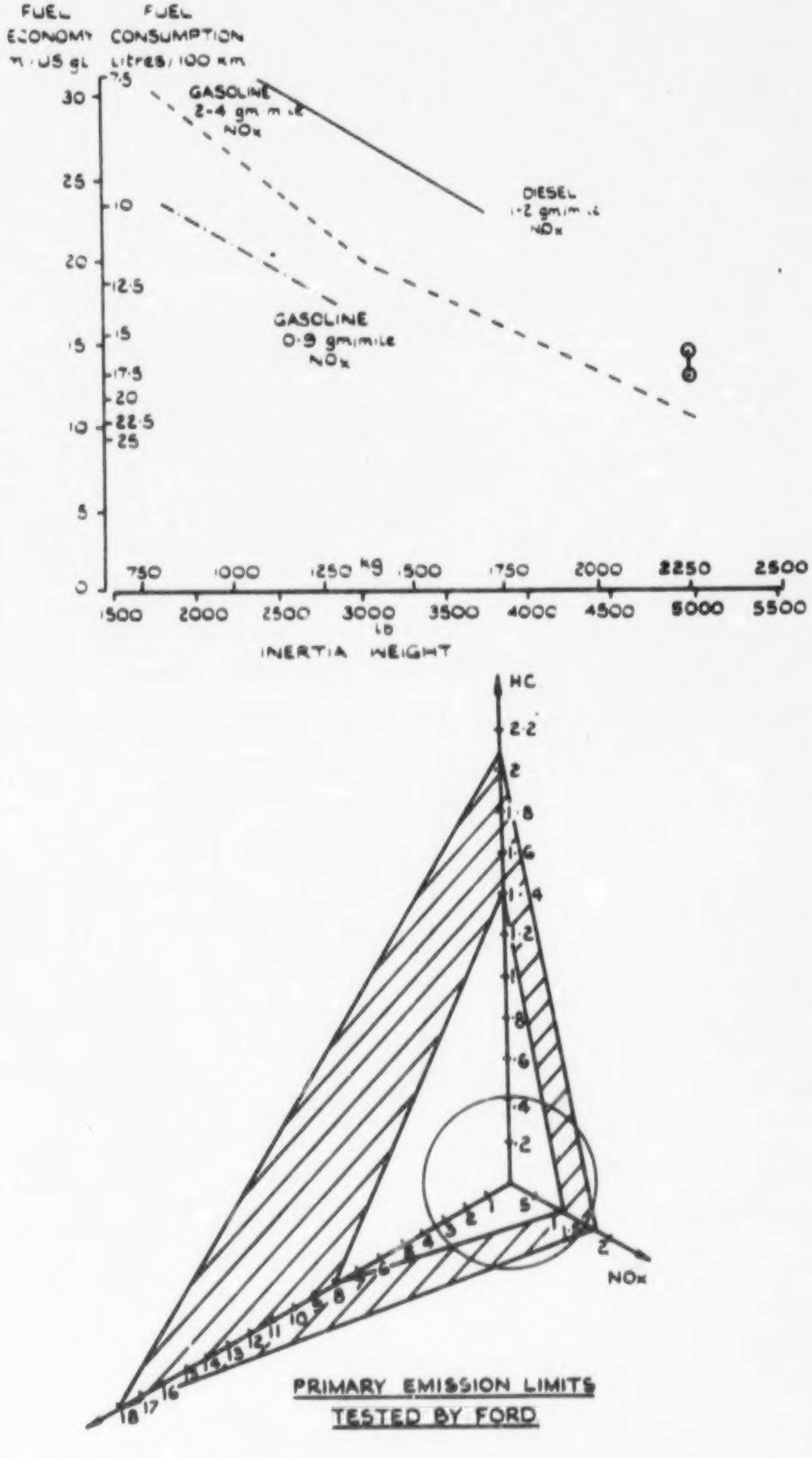


Figure 1.9. Emissions and fuel economy for 400 CID Proco vehicles 110T606, 110T612, 311 to 42 automatic transmission.



PRIMARY EMISSION LIMITS

TESTED BY FORD

Figure 1.10. Emission results from M151 Proco vehicles without catalysts. Transmission: Manual.

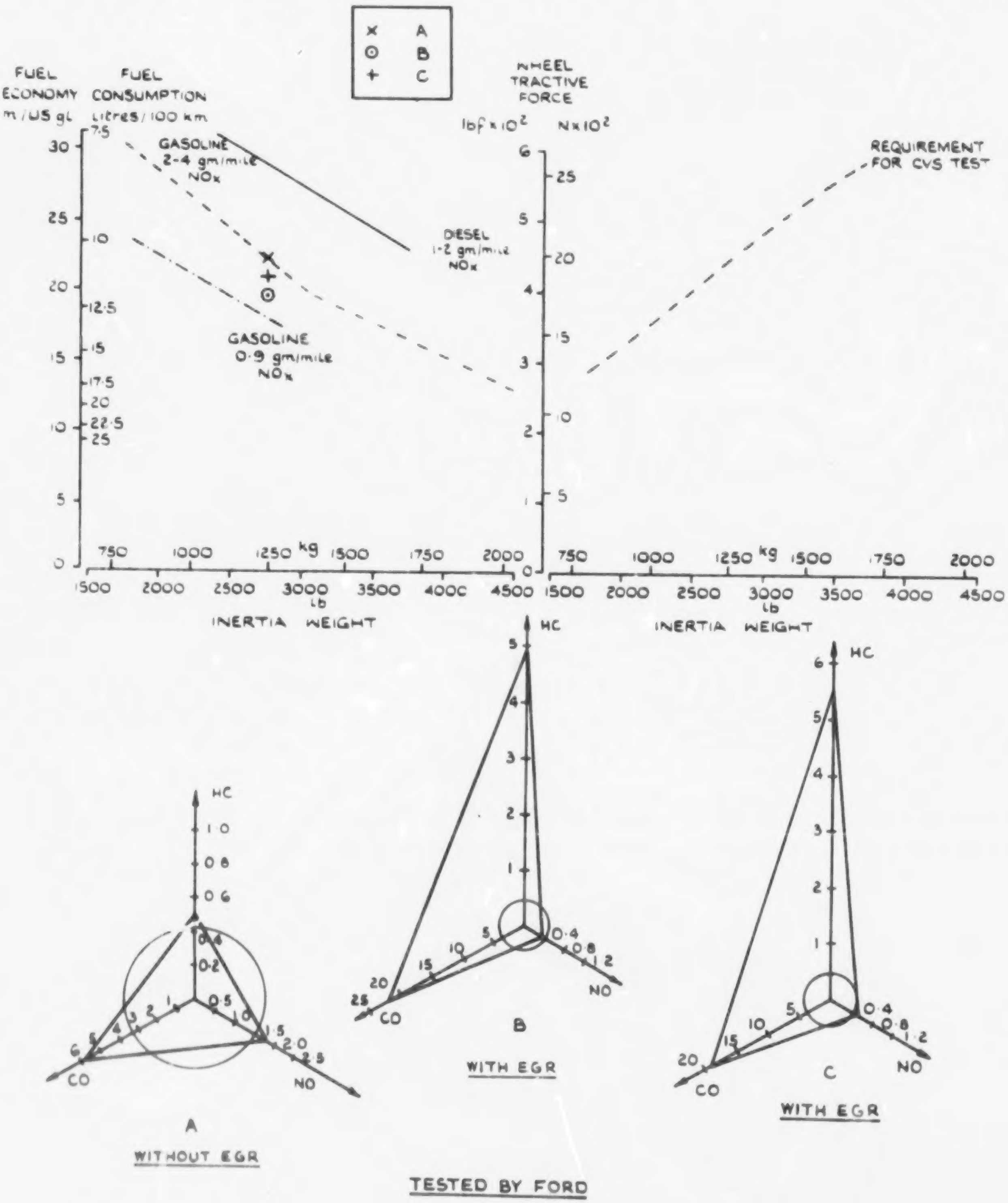
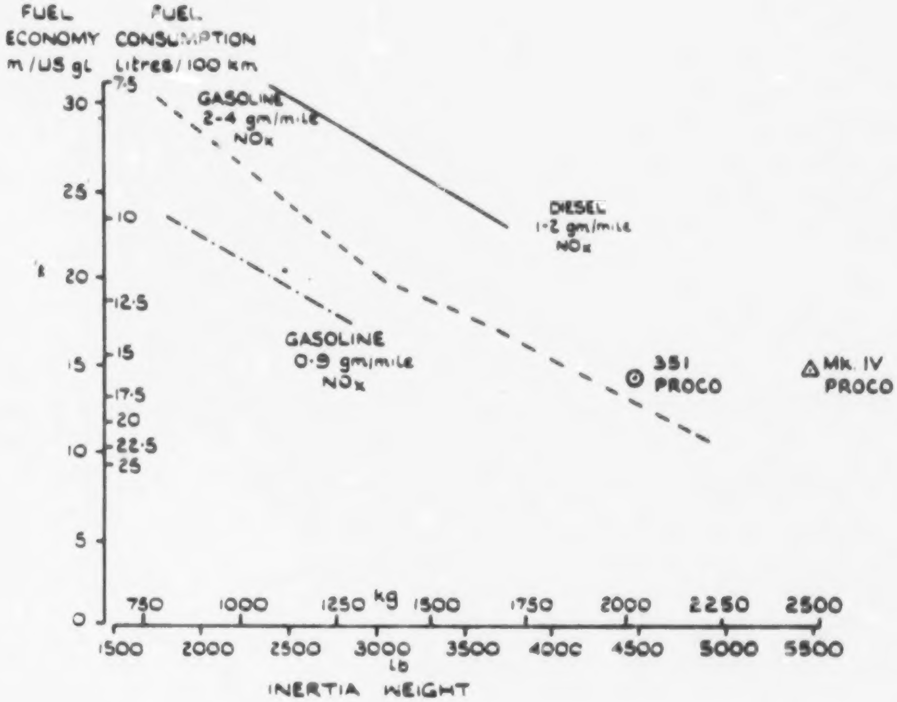


Figure 1.11. Emissions and fuel economy of super economy Mk IV Proco and 351 durability Proco NO emission controls.



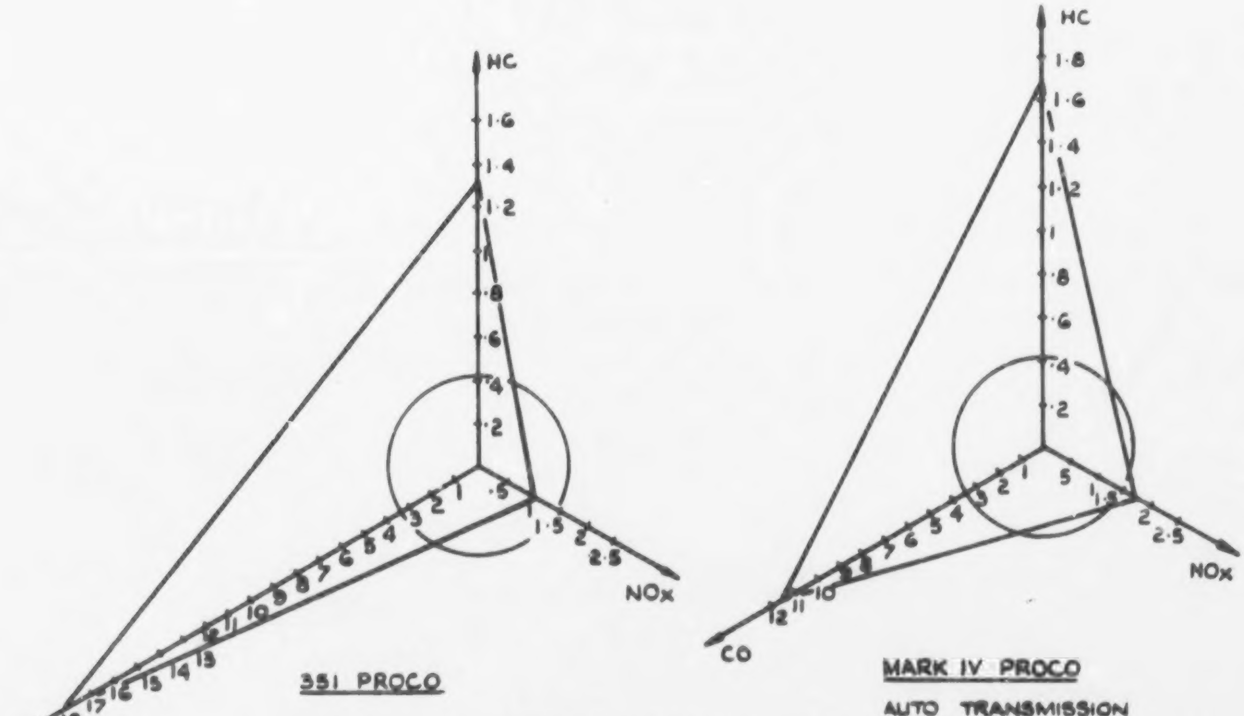


Figure 1.12. Comparison of emissions and fuel consumption of diesel engine and Ford FCP engine (data extrapolated from single cylinder).

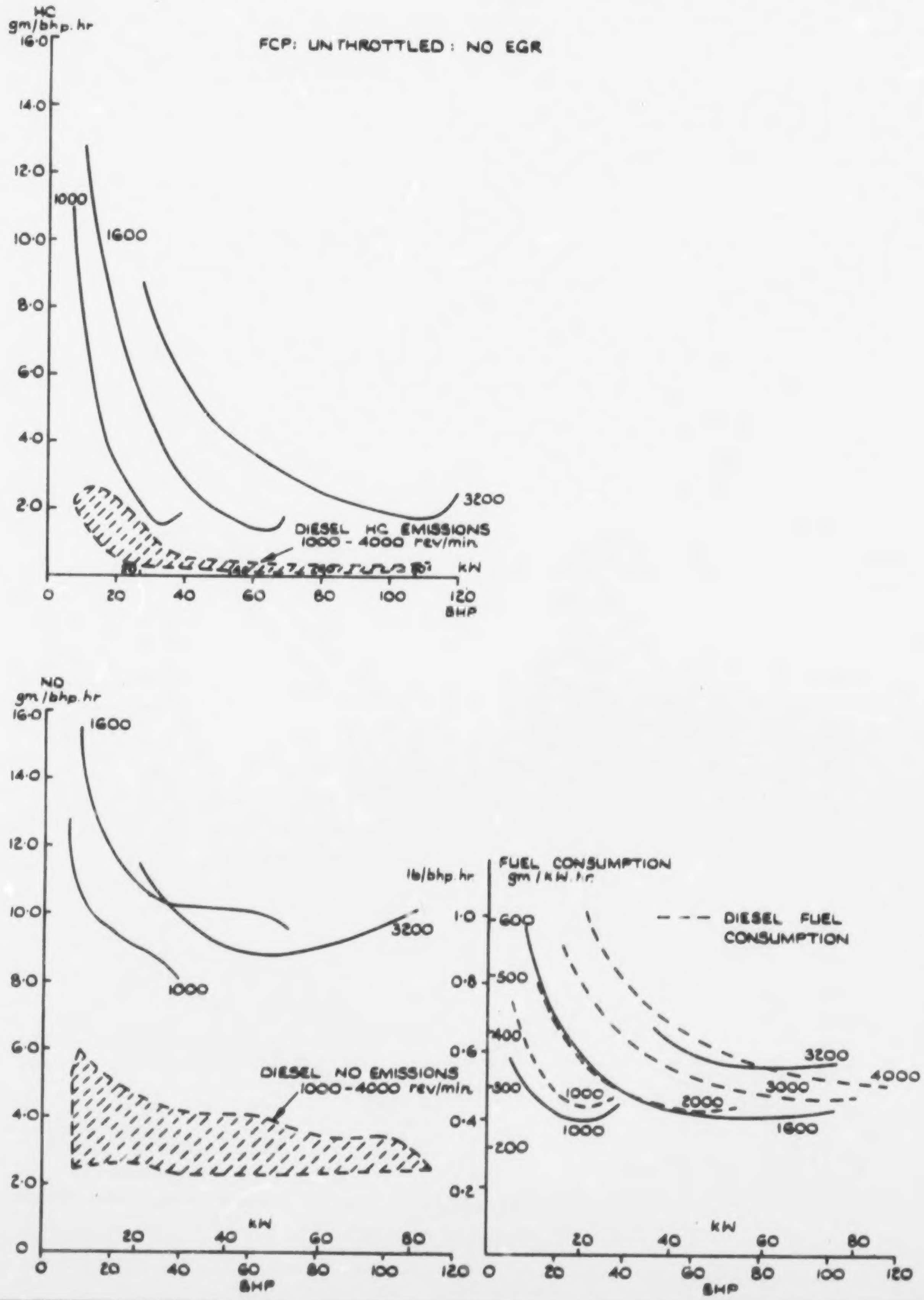


Figure 1.13. Comparison of emissions and fuel consumption of diesel engine and Ford Proco engine (data extrapolated from L141 engine).

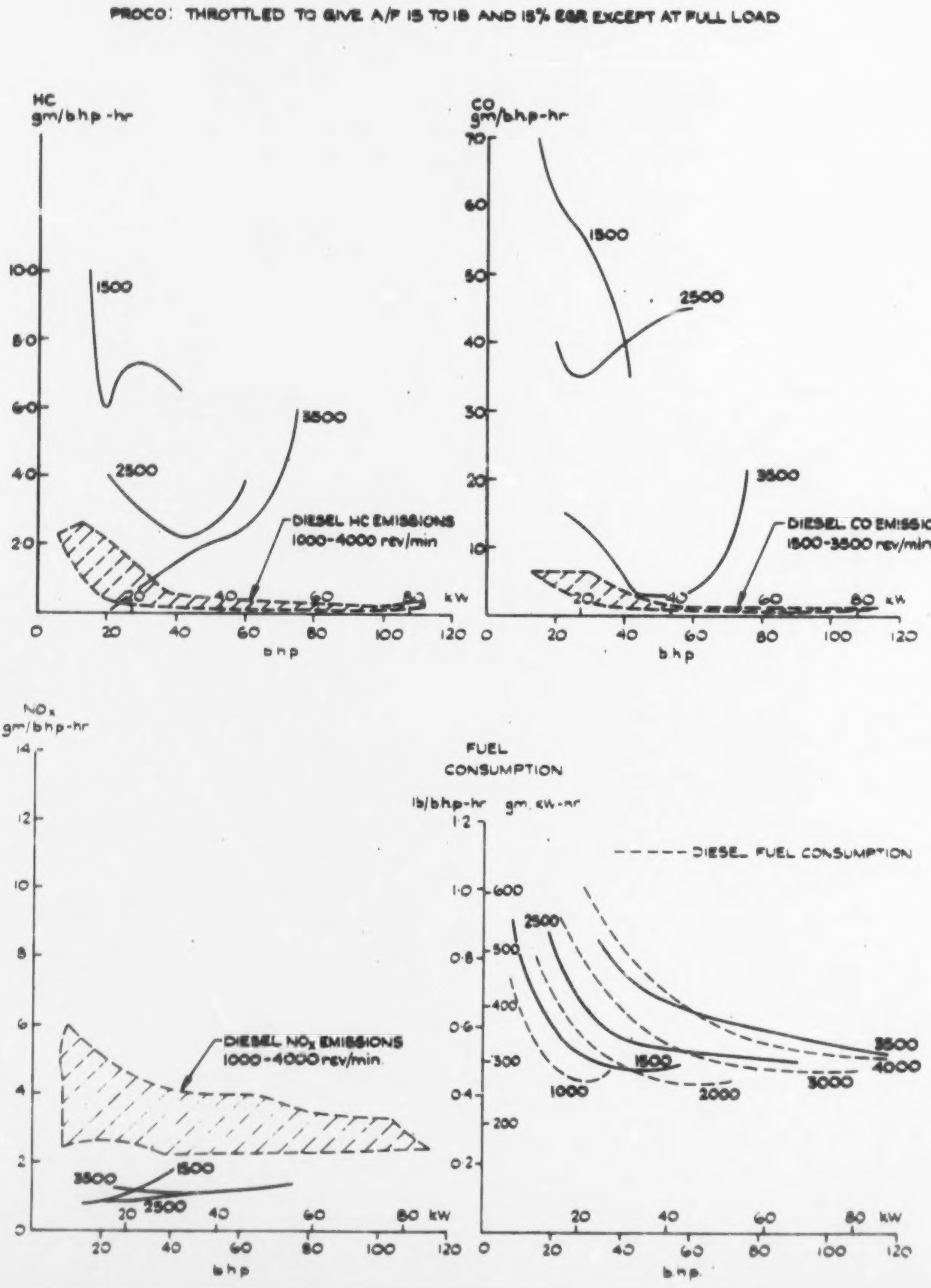


Figure 1.14. Comparison of emissions and fuel consumption of diesel engine and MCP engine (data scaled from single cylinder).

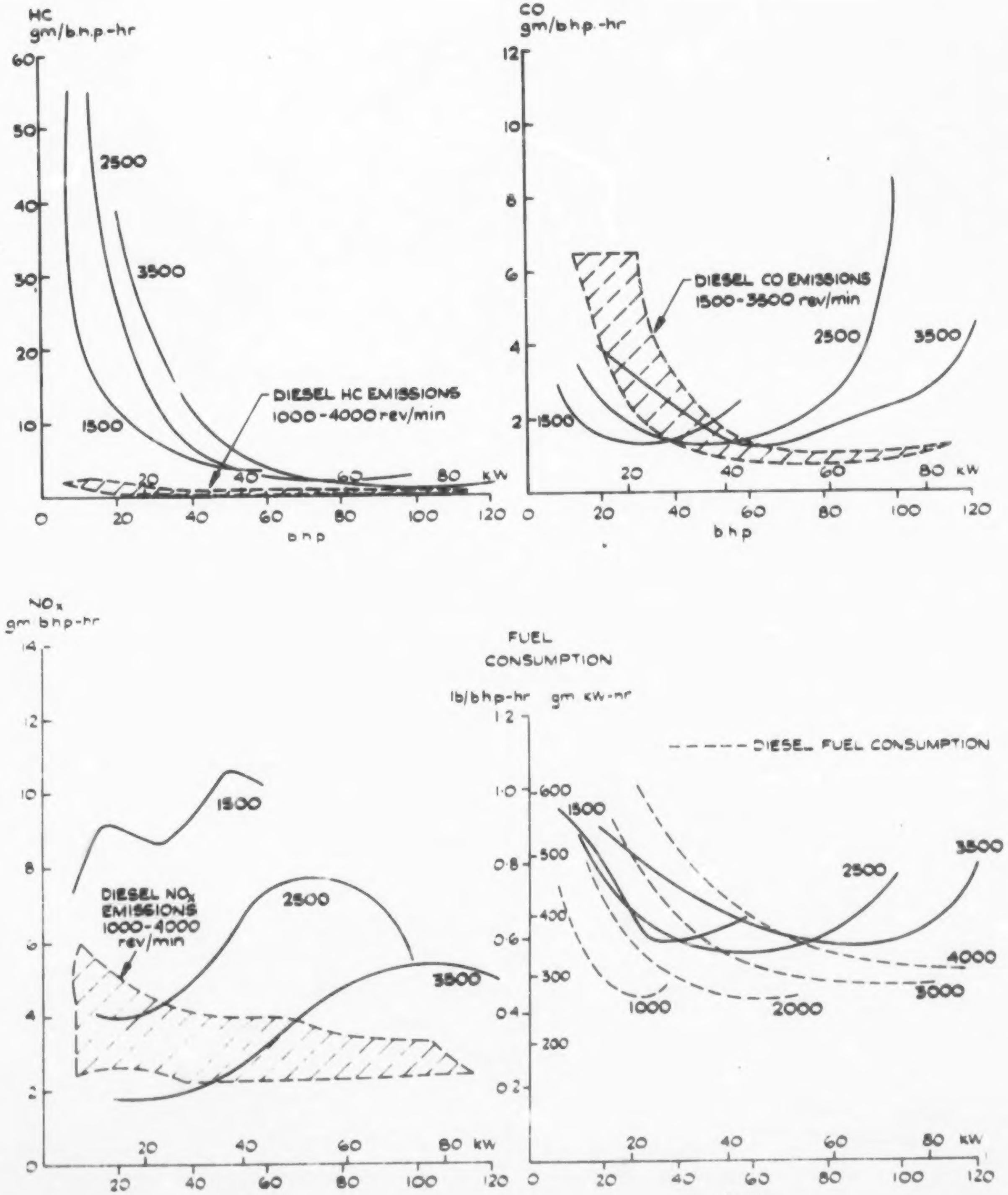
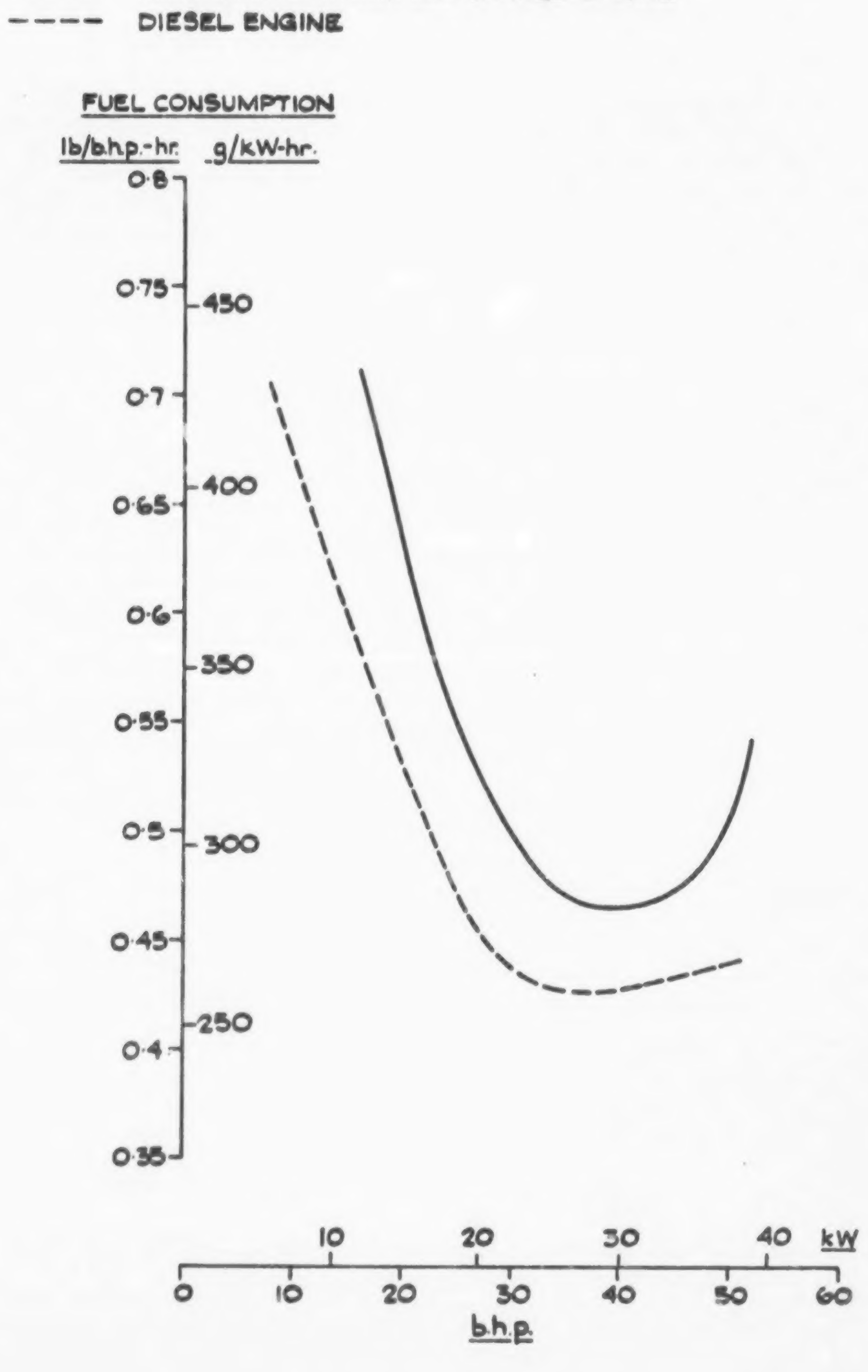


Figure 1.15. Comparison between diesel engine and Hesselman stratified charge engine @ 1500 rev/min.



CATEGORY II

SINGLE COMBUSTION CHAMBER WITH FUEL INJECTION DURING COMBUSTION: MIXTURE FORMED BY FUEL INJECTION AND/OR AIR MOVEMENT

History

The Texaco combustion system has become the best known stratified charge engine, due to the considerable development effort which has been expended on the process over the last 30 years. The basic principle is that combustion can be initiated by a spark plug and then controlled by the rate of the fuel injection. Two other companies have followed the same principle, Deutz who have a similar arrangement to that of Texaco, and Curtiss-Wright who applied the system to a rotary engine.

Topic Area (a) Combustion Chamber and Stratification Technique

As in Category I, this topic area will be sub-divided so that each of the design factors which control the combustion can be inspected separately. First, the general combustion chamber arrangement will be described.

In the Texaco process, ref. 2.1, high intensity air swirl is created by the induction port and a masked valve and this swirl level is magnified at TDC by a cup-in-piston arrangement (Fig. 2.1). Some fuel is injected, which mixes with the swirling air and then passes a spark plug which is positioned beside the injector. The spark plug ignites this initial mixture, and a burning zone is produced between injector and spark plug. The combustion is then controlled by the rate at which fresh fuel is fed to the burning zone. However, the fuel flow rate must be matched to the air swirl level, which is also entering the burning gases. If air and fuel flow are not matched, then the flame may go out, or carbon particles may be produced and exhausted as smoke. Engine torque output is controlled by the amount of injected fuel, and this also affects the duration of combustion. The TCCS (Texaco Controlled Combustion System) has been applied to a wide variety of engines ranging in sizes from 83 mm (3-1/4'') to 147 mm (5¾'') bore in both single cylinder and multi-cylinder forms.

The Deutz process resembles TCCS in principle, ref. 2.3, but the piston cavity is angled from the cylinder centreline, to simplify installation of the spark plug and injector (Fig. 2.1). This combustion process also depends on induction air swirl and rate of fuel injection. The Deutz stratified charge engines were not developed specifically for this purpose, but rather are an adaption to an existing range of diesel engines. The function of the spark plug is to give the engine some multi-fuel capability. This could also be achieved by increasing the engine compression ratio, but this approach has certain disadvantages with respect to noise and specific power output. The Deutz system has been applied mainly to bore sizes of 120 mm (4.7'') in engines with as many as 12 cylinders. This is one of the few stratified charge engines in production and they are employed in truck and military applications.

Since high levels of air swirl are generated by the induction processes of these engines, volumetric efficiency at high speed is necessarily limited. Applying the same combustion principle to a Wankel rotary engine avoids this problem, since the air charge has a high velocity at TDC, due to the mechanical configuration of the engine (Fig. 2.2). However, other critical design areas are introduced. For example, the injector and spark plug must not interfere with the rotor seal, or cause excessive gas leakage as the seal passes over the recesses. The rotor cavity of the rotary engine is often specifically shaped to improve fuel air mixing at the time of combustion. Curtiss-Wright have worked on this approach applying it mainly to large engine sizes in single and dual bank configurations, ref. 2.2.

Texaco, Deutz and Curtiss-Wright all claim similar advantages for their injection controlled combustion processes, i.e. a multi-fuel engine where the fuel does not have octane or cetane requirements, an ability to use a variety of compression ratios, and unthrottled operation with the associated fuel consumption benefits.

Injectors and Spark Plugs.

The development of the layout of spark plug and injector have followed remarkably similar paths in the Texaco and Curtiss-Wright engines. In early versions of TCCS and Curtiss-Wright, the spark plug was positioned remotely from the injector. For TCCS engines, the included angle between spark plug and injector about the bore centreline was 30-80°. In some of the Curtiss-Wright engines they had trenches connecting the spark plug and injector recesses. With development, all three engines in this category appeared to favour locating the spark plug on the edge of the fuel spray, and as close to the injector as possible.

The actual types of spark plug and injector have been tailored to each combustion system, and in some cases do not follow either conventional gasoline or diesel engine practice. The literature surveyed indicates that the injector in the TCCS engine is of the inward opening pintle type, except that the sealing end of the pintle is flat, Fig. 2.3. The theory behind this arrangement is that the high pressure fuel jet experiences a high shearing action at the sharp edges, and a fine spray is produced on the edge of the main fuel spray. This fine spray vaporises quickly and can be more easily ignited than the main fuel stream. Work is currently being carried out to determine the suitability of more conventional nozzles for the TCCS system. The fuel injection pump is usually of the distributor type which has cost advantages over the in-line pump type. Curtiss-Wright and Deutz apparently use conventional injector types, Deutz using single and dual hole offset sprays, and Curtiss-Wright with dual and multiple hole types. These injectors can have a disadvantage with relatively large sac volumes between the controlling needle and the hole, which can contribute to engine hydrocarbon emissions. This problem would be less severe with a rotary engine, where the injector does not experience atmospheric pressure during the working cycle, and evaporated fuel may not necessarily escape from the sac. Deutz, Curtiss-Wright and TCCS engines all use high pressure diesel fuel injection equipment, with nozzle opening pressures in the region of 100-200 bar (1500-3000 lb/in^2).

The spark plug in the TCCS is positioned such that the earth electrode is perpendicular to the fuel spray. This orientation is beneficial when operating on heavier fuels, while any orientation may be employed with gasoline. If angular orientation of the plug is important, the normal body threads cannot be used, and the plug must be located by a retaining sleeve. Contradicting this approach, Deutz developed a special spark plug where the high tension electrode was masked from the direct fuel spray; there were windows cut in the sides of a protecting closed cylinder, to allow the initiating flame to spread into the combustion chamber (Fig. 2.4). Curtiss-Wright have apparently experimented with a variety of spark plugs and injector geometries, including a close coupled arrangement where the injector forms the earth electrode for the plug. In another promising build, there were two injectors to each spark plug, one to produce a pilot spray which the plug ignited, and the other to deliver the main fuel spray. Another build was a twin arrangement of two spark plugs and two injectors with one set on each side of the longitudinal housing centreline, Fig. 2.5, which gave a wide operating spectrum. This also improved air utilisation which is a problem area with this type of stratified charge engine.

Ignition.

A long duration multi-strike is preferable, so that sooner or later, when ignitable mixture occurs at the plug, that mixture can be fired. These requirements indicate that the mixture is far from homogeneous at the plug, and the edge of the fuel spray may simply be partially evaporated fuel droplets. Alternatively, some of the ignition energy may actually be used in evaporating the fuel, so that combustion can begin. TTIS (Texaco Transistorised Ignition System) is a special ultra-high energy ignition system which has been developed specifically for the TCCS process, and has also been used with some success in the Curtiss-Wright engines. The energy delivered by TTIS is in the region of 30 millijoules per spark discharge in comparison with about 10 millijoules for a conventional coil ignition. The higher energies are not produced by higher voltages but from a different voltage and current waveform, and a longer total ignition duration. (TTIS and conventional coil ignition breakdown voltages in an operating gasoline engine = 8-10 kV).

The ignition system of the Deutz engines resembles TTIS, in that it has a multi-strike capability; 2-20 sparks occur at a rate of about 1 per crank degree, depending on the spark gap. However, the high engine compression ratio and pressure boosting apparently reduces the breakdown voltages to the region of 5-15 kV. Voltages around 25 kV are needed for cold starting.

Compression Ratio.

The following compression ratios have been used:

TCCS Naturally Aspirated = 10
TCCS Turbocharged = 9.3 and 10
Deutz Turbocharged = 16
Curtiss-Wright = 8.5 and 9.5

The large variations in the compression ratio of the similar Texaco and Deutz processes is probably a reflection of the different backgrounds of the experimenters; Texaco attempting to improve the gasoline engine, and Deutz the diesel engine. As compression ratio increases, there are three conflicting factors affecting fuel consumption. On the one hand, the fuel energy is used more efficiently, and on the other, maximum power decreases, and mechanical friction increases. An approximate optimum compression ratio figure obtained from diesel engine practice is 14, and this will probably also apply to stratified charge engines. However, another important factor influenced by compression ratio is hydrocarbon emissions, which tend to decrease as compression ratio increases in heterogeneous combustion. Therefore, on balance, the Deutz compression ratio of 16 is probably closer to the ideal than Texaco.

The compression ratio of the Curtiss-Wright engines appears unusually low, but there are two practical difficulties. As already mentioned, the recesses for the spark plug and injector cause leakage of burnt gases back into the trailing lobe as the apex seal passes the recesses. Hot particles of burnt material can cause pre-ignition of the first unit of injected fuel. This effect was observed by Curtiss-Wright while running on diesel fuel when the spark plug and injector recesses were larger than usual. This observation is interesting, in that it demonstrates that there must be a significant delay between injection and combustion. The other limitation on compression ratio in the Curtiss-Wright is purely geometrical. At the particular rotor/eccentricity ratio used, the maximum compression ratio is 17, with the rotor cavity volume reduced to zero.

Swirl.

The design of TCCS engines is completely dominated by the very high swirl requirements. Charge rotation is generated during the combustion process, using a directed port, or a masked valve as in the early designs. This charge swirl is then intensified at TDC by a deep piston bowl.

Ricardo measurements indicate the following swirl rates on Deutz and Texaco ports at the end of the induction stroke.

TCCS = 3.5 x engine speed

Deutz = 3.0 x engine speed

Therefore, other factors being equal, the Deutz engine would be expected to give better high speed volumetric efficiency. Piston cup sizes taken from drawings are tabulated below although they are influenced to some extent by the engine bore size.

TCCS piston cup = 50-58% of bore dia

Deutz piston cup = 39% of bore (120 mm bore)

These figures would indicate that the final swirl level for the two engines at combustion is probably very similar. Texaco engines have final swirl levels in the region of 7-10 x engine speed. The Texaco engine piston cup size cannot easily be reduced any further due to interference of the injector and spark plug with the ports. The beauty of the Deutz process is in the inclined cup, which prevents this interference, and allows a smaller piston cup to be used

Both Deutz and Texaco publications mention an upper limit on gas velocity in the piston cup of 90 m/sec. Above this figure, the gas velocity has a significant effect on the injector spray characteristics and ignition. A simple calculation indicates that for an engine operating up to 4500 rev/min, with a final swirl ratio of 8.5, the maximum piston cup diameter is 45 mm.

The rotary engine has an inherent gas velocity past the injector, which is determined by the rotor cavity dimensions and engine size. These dimensions do not compromise the engine volumetric efficiency which is usually higher than that of reciprocating engines.

Throttling.

Generally, the engines in this category are not throttled, although some TCCS engines have been throttled at idle, light load and decelerations. This controls air/fuel ratios to below 60:1.

Topic Area (b) Combustion Characteristics

With conventional injection equipment it is impossible to burn effectively the first unit of fuel coming from the nozzle, and then control combustion by injection. Time must be allowed for fuel to evaporate, therefore the combustion is controlled by evaporation rate rather than injection. Texaco quoted figures suggest that the evaporation delay time is 6°, if injection is in the direction of the air stream and 10-20° if injection is against the air stream. A cumulative mass burn curve for the TCCS process suggests that the delay time is 12°, Fig. 2.6, and that over 40% of the fuel has been injected by the time combustion commences.

The period of combustion is an important parameter, as it measures how close the combustion approximates to constant volume conditions. In a good gasoline engine, the time between 10-90% of the fuel being burnt is typically about 30° of crank rotation. The TCCS process apparently requires about 55° at full load, Fig. 2.6, although this will be reduced at low loads. Long duration full load combustion appears to be an inherent characteristic of the processes in this category. In the Curtiss-Wright rotary engines combustion was 90% complete at 40° ATDC (total time about 60°) for the dual injector arrangement, and the corresponding figure for the single injector was 60° ATDC (total time about 90°). These figures for the rotary engine should be multiplied by two thirds to achieve a comparison with the reciprocating engine.

An important advantage of this category of stratified charge engine, over those in Category I is a smooth and stable cylinder pressure diagram, Fig. 2.7, which leads to low noise and reduced engine stresses. This probably arises from the fact that only a small amount of fuel and air mixing occurs before burning commences. Cyclic variations in cylinder pressure appear to be much lower than for gasoline engines.

Details on the heat rejection of this category of engine are absent from the literature. However, one would certainly expect heat losses to be higher than those of a gasoline engine at the same load. This is due to forced convection from the high intensity swirl or gas movement in the case of the rotary engine, and radiation heat transfer from carbon particles in the rich region of the combustion chamber. At low loads, the position will probably be reversed, and the gasoline engine would have greater heat loss.

Topic Area (c) Performance

Due to the different sizes and applications of the Category II engines in the published literature, it is difficult to make a simple performance comparison between them. The principal factors affecting performance are listed below.

1. Volumetric Efficiency. The rotary engine can attain volumetric efficiencies better than 100% with peripheral ports due to good flow characteristics of the ports and unsteady gas pulsing effects. Power output in a Curtiss-Wright engine has been increased by 16% by changing from side to peripheral ports. This potential cannot be exploited so easily in a conventional throttled gasoline rotary engine, due to the timing arrangement of peripheral ports which increases the amount of charge residuals trapped in the chamber. The TCCS engine has a good volumetric efficiency at low speed (circa 87%), but this is compromised at high speeds by the air swirl requirements. Deutz and Texaco both use turbocharging to compensate for this poor high speed volumetric efficiency.

2. Maximum Speed. In spite of the relatively large swept volume, the rotary stratified charge engine operates to 5000 rev/min. Power output would be further increased if this maximum speed were extended, although new fuel injection equipment for these speeds would be required. TCCS engines all seem to reach their maximum power points at 3000-3700 rev/min. In the case of the L141 engine, a limit stop operated at that speed. The literature refers to "mechanical reasons" for this limitation. Since the volumetric efficiency of this engine is decreasing rapidly above 3000 rev/min, it is unlikely that the maximum power point could be extended above 4000 rev/min. Since the Deutz engines have been developed for large trucks, high engine speeds are less important, and the published graphs only cover the speed range to 2800 rev/min.

3. Smoke Level. Although the power of all the engines in this category is limited by the exhaust smoke level, neither Curtiss-Wright nor Texaco have included smoke levels with their published power curves. The information used in the TCCS graphs in this section has been obtained privately from Texaco. In TCCS engines a Bosch smoke level of 2 occurs at an air/fuel ratio of 18-22:1. Air utilisation in the Curtiss-Wright engines appears to be even poorer, and the performance curves only cover a fuelling range to 20-25:1 air/fuel ratio. This tendency for the stratified charge rotary engines to produce smoke is compensated by the high volumetric efficiency.

Test Bed Results.

A torque curve extrapolated from the rotary RC2-60 is shown in Fig. 2.8 The abscissa of the rotary engine curve has been adjusted to coincide with the speed range of the baseline diesel engine. The swept volume has been calculated taking into account all three lobes of a rotor. Swept volume figures cannot be compared meaningfully between rotary and reciprocating engines, and a complete configuration study would be required to determine overall engine sizes. The fall in engine torque as speed is reduced is not wholly a function of port timing, but is also affected by seal leakage. At lower speeds leakage has a considerable effect on both torque and fuel consumption.

Due to the low maximum speed of the Deutz and TCCS engines, it was unrealistic to compare them with the baseline diesel on a horsepower basis so the BMEP was used as the graph ordinate. The specific torque output of naturally aspirated TCCS engines is shown in figs. 2.9 and 2.10 and when compared with typical gasoline engines is low, although similar to current diesel engines. The single cylinder curve was adjusted to the friction level of the L141. At the same smoke limit, gasoline fuel yields 10-20% greater output than diesel. In a turbocharged build the L141 gave good high speed torque but relatively poor torque at low speeds. The boost density ratio was 1.08 at 1800 rev/min. and 1.26 at 3000 rev/min. In the SWRI results on a turbocharged Cricket, the boost density was 1.3 at 1800 rev/min. and 1.4 at 3000 rev/min. On diesel fuel this engine gave higher torque than on gasoline, but smoke records on CVS tests indicate that the smoke level was significantly higher than for gasoline. The fuel pump stop had obviously been set for gasoline, so that a greater weight of diesel fuel was delivered due to its higher specific gravity. Deutz also observed that on the same rack position, diesel fuel produced 15% more torque than gasoline.

There was no convenient data to make performance comparisons with the turbocharged and intercooled Deutz engine, as the boost density ratio was not specified. The graph in Fig. 2.11 compares the stratified charge version of the engine with its diesel counterpart. At the same smoke level, there was a marginal increase in torque for the stratified charge engine operating on gasoline, compared to the diesel.

Vehicle Results.

The L141 engine has been installed in 2 cars, an M151 jeep and a Cricket sub-compact. In an uncontrolled emissions build, the L141 TCCS engine powering a 1250 kg. jeep would follow the most demanding accelerations of the CVS-CH test. However, to achieve very low NO_x levels, there is a penalty in power output. At 0.4 gm/mile NO_x, the naturally aspirated M151 jeep would only just follow the least demanding acceleration of the driving cycle, and could not attain a speed of 55 mile/h. Estimated wheel tractive force at 33 mile/h for the M151 is shown in Figs. 2.12 - 2.14. The figures were calculated from the engine full load torque curve at Bosch number 2 and the transmission ratios. It was assumed that the engine suffered a 30% loss in torque output in meeting the secondary emission targets. (Taken from Texaco published figures).

Tractive force for the Cricket could not be calculated, as the characteristics of the torque converter in the automatic transmission were not known.

In conclusion, it can be stated that specific power output of the engines in this category is generally low due to smoke limited specific torque, and a restricted maximum speed range for the reciprocating engine. Matching the performance of conventional gasoline engines would require either a considerable increase in engine size, or turbocharging. Both of these approaches would probably be necessary to meet the secondary emission target without a performance penalty.

Topic Area (d) Fuel Consumption

Friction.

In theory unthrottled stratified charge engines should have better fuel consumption than gasoline and diesel engines, due to lower frictional losses than the diesel engine, and better cycle efficiency than the gasoline engine. The TCCS system exploits this potential for low friction, since conventional production gasoline engines can be adapted to the process, due to the low cylinder peak pressures. Fig. 2.16 shows frictional values for three 2.3 litre (141 in^3) engines and a single 3.0 litre (183 in^3) engine. The reason for the relatively low values at 2500 and 3000 rev/min. of the L141 is not known. The low friction of unthrottled engines is particularly noticeable at idle where only 5 mm^3 of fuel is required per working cycle (TCCS). This compares with 10 mm^3 for diesel engines and throttled gasoline of comparable sizes. Some authorities consider idle fuel consumption to be important for automotive applications such as taxi service. Since the Deutz process has been applied to an existing diesel engine structure, there would not be a difference in motoring friction, since the compression ratio was not changed.

There is a greater variation in motoring friction between types of rotary engines than of reciprocating engines, due to the effect of sealing efficiency. In-house results from a Mazda RX2 and published figures from the NSU KKM 612 are shown in Fig. 2.17. Also plotted is a motoring curve for a Curtiss-Wright RC2 60U5 carburetted rotary engine, which formed the basis for the stratified charge version. The rev/min scale of the rotary engines have been adjusted to align peak power speeds, assuming that peak power occurs at 5000 rev/min for the rotary engine (this corresponds to the Curtiss-Wright stratified charge engines) and 4000 rev/min for reciprocating engines. The high frictional levels of the Mazda engine were the subject of a Ricardo study, which involved evacuating the engine with a vacuum pump to separate pumping and heat loss terms from pure mechanical friction. It was shown that the high el of this rotary engine was solely due to pumping and heat loss terms, Fig. 2.18, and a computer model indicated that seal leakage was probably the major component.

The lower leakage level of the NSU and Curtiss-Wright engines was attributed to the greater care taken in the construction and assembly of prototype engines. The motoring friction for the Curtiss-Wright stratified charge engine would be expected to be higher than the carburetted counterpart due to the extra gas leakage caused by the injector trenches in the housing.

Test Bed Results.

A fuel consumption map from an RC1-60 rotary engine was used to estimate the fuel consumption performance of a 96 kW (128 hp) version. A smoke limited output of 6.2 bar (90 lbf/in^2) was assumed at a speed of 5000 rev/min. Fuel consumption was good at higher speeds, Fig. 2.19 but the efficiency obviously suffered at lower speeds due to leakage. In Fig. 2.19 a rotary engine speed of 5000 rev/min should be compared with a diesel engine speed of 4000 rev/min, and this ratio should be retained over the speed range. Curtiss-Wright figures of motoring and firing engine air flows show that seal leakage in this engine is appreciable. A single fuel consumption curve taken from an RC1-90 and extrapolated to 96 kW is shown in Fig. 2.20, and is slightly inferior to the RC1-60 consumption.

The TCCS engines give low test bed fuel consumption figures at low loads, as illustrated by results from L141 and LIS 183 engines, Figs. 2.21 and 2.22. However, above two-thirds load,specific brake consumption increases, rather than decreasing or remaining flat in the manner of gasoline or diesel engines. This characteristic indicates inefficient combustion, and high smoke and CO emissions in this region would suggest that the fuel is not completely oxidised. Turbocharging the engine decreases the mixture strength and specific fuel consumption then remains flat. The low load fuel consumption is not affected by turbocharging. In addition, specific fuel consumption on the L141 was not affected by the type of fuel, whether the engine was naturally aspirated or turbocharged. However, the LIS 183 gave a much inferior full load consumption on diesel fuel, Fig. 2.22, at the present stage of development. Further optimisation is probably required.

Published results from the Deutz engine extend over a limited torque range and are difficult to compare with naturally aspirated engines, due to the high boost, Fig. 2.21. Since friction levels are identical to the D.I. diesel counterpart, there are unlikely to be fuel consumption gains. A single fuel consumption point taken from a single cylinder naturally aspirated Deutz engine, and extrapolated to the target horse power, corresponds with the diesel engine baseline, Fig. 2.23.

Some TCCS engines have been lightly throttled at low loads to maintain higher exhaust temperatures for emission control purposes. Single cylinder tests have shown that maintaining an air/fuel ratio of about 40 at loads below 4 bar (60 lbf/ in^2) IMEP has only a very small effect on fuel consumption.

This observation corresponds with thermodynamic predictions.

Vehicle Results.

Vehicle fuel economy would be expected to benefit from the excellent low load consumption of the TCCS engine. However, on the CVS-CH tests, the gains are relatively minor and do not meet the diesel baseline. At NO_x levels of 1.5 g/mile, turbocharged Cricket and M151 vehicles operating on gasoline, gave fuel economies as good as and slightly better than uncontrolled gasoline engines, Figs. 2.12, 2.13, 2.15. Meeting the secondary emission target of 0.4 g/mile NO_x involved a considerable economy sacrifice in both turbocharged and naturally aspirated forms, Figs. 2.14 and 2.12. The relationship between economy and NO_x level is apparently almost linear, Fig. 2.24. A sub-compact car suffered 20% loss in fuel economy and an M152 suffered 33% loss in meeting 0.4 g/mile NO_x. Vehicle fuel economy figures were not available for the Deutz and Curtiss-Wright processes.

In Fig. 2.15 fuel economy is superior on heavier fuels. The difference can be attributed to the specific gravity of the fuel, Fig. 2.25. However, in the case of the M151, Fig. 2.14, there is a discrepancy between the fuels even when specific gravity is taken into account, Fig. 2.26.

The apparent contradiction between test bed and vehicle fuel consumption results on the TCCS justified further study. The test bed figures were significantly lower than the baseline diesel engine at low loads yet the vehicle fuel economy was worse. It has already been established that the CVS driving cycle has a relatively low load factor, so some gain in fuel consumption over the baseline diesel, should be apparent. One possible explanation is that, in the vehicle, the engine operating parameters were re-set from optimum fuel economy, to reduce exhaust emissions. Calculations were performed to check the fraction of fuel being emitted as exhaust hydrocarbons, but this only affected fuel consumption by 2.5%. It is possible that the vehicles adapted to the TCCS processes have relatively poor transmission efficiencies compared to the baseline data, and this might explain the anomaly in fuel consumption. The transmission efficiency of the M151 jeep may not be typical of passenger cars. The other vehicle on the graphs, is a TCCS Cricket which has automatic transmission, therefore fuel economy would be expected to be worse than the manual transmission vehicles which were used to construct the gasoline and diesel baselines. The choice of automatic transmission probably masked the poor driveability which would have resulted from a manual transmission. The naturally aspirated TCCS process has very poor torque output and therefore must be boosted but turbochargers are not effective at low engine speeds so initial vehicle accelerations are poor. Using an automatic transmission together with the turbocharger increases the engine speed at low vehicle speeds and significantly improves vehicle performance. Therefore, the overall conclusion from this topic area is that the stratified charge engines in this category have displayed excellent test bed fuel consumptions, which have not been reflected in the vehicle, possibly due to the transmission characteristics of the vehicles chosen.

Topic Area (e) Emissions

HC, CO and NO_x from Engine Test Beds.

The literature search only uncovered a limited amount of uncontrolled exhaust emission data from engine test beds in this category. An emissions map was available for a naturally aspirated L141 TCCS engine, and this was adjusted to the target horsepower, but there were no details on emissions from turbocharged versions. An emission curve has been published for a Curtiss-Wright stratified charge engine, and only a single operating condition for a naturally aspirated Deutz engine.

In common with Category I, hydrocarbon emissions from Category II engines are higher than conventional gasoline engines. This particularly applies to the TCCS process where low load hydrocarbons may represent over 2½% of the fuel consumed, Fig. 2.27. The Curtiss-Wright engine emits a similar level, Fig. 2.20, but a single point from the naturally aspirated single cylinder Deutz engine appeared to be lower, Fig. 2.23. Carbon monoxide emissions are lower than from gasoline engines at light and full load, but are higher in the part load range. Since excess air is always present in the combustion chamber of stratified charge engines, the high CO concentration levels of 0.1 to 1% are indicative of poor mixing after combustion. Therefore, in spite of the vigorous air motion in the Curtiss-Wright and TCCS engines, mixing and combustion are not completed. Nitric oxide emissions are lower than gasoline engines, and in the case of TCCS and Curtiss-Wright, are comparable with the diesel. It is interesting that the form of the NO_x emission curves with load, do not resemble any other engines, except for the MCP in Category I.

A very important aspect of the TCCS engine, which probably applied to the other processes in this category, is a tolerance to high EGR levels, resulting in the ability to meet very low NO_x emission standards. 15% EGR gave an 80% reduction in NO_x in the L141 engine and only marginally increased engine hydrocarbons. However, indicated specific fuel consumption suffered by 30% and there was a 75% increase in exhaust temperatures. These factors point to a considerable deceleration in the rate of engine combustion as EGR is added.

HC, CO and NO_x from Vehicles.

There is no published information on the application of Curtiss-Wright or Deutz stratified charge engines to automotive applications. Vehicle emission results for TCCS cars without emission controls correlate well with the observed test bed results. Hydrocarbon levels are high, around 2-3 g/mile, while CO levels lie between conventional gasoline and diesel engines, Fig. 2.13. NO_x levels are comparable with the diesel baseline.

Reducing HC levels to within the primary emission targets of 0.41 g/mile requires over 85% control of the TCCS emissions. A typical platinum based catalytic reactor will yield 90% control when new and 70% after an extended period of driving. For this reason, Texaco incorporated three emission control devices in the exhaust of some TCCS engines; the first was a swirl reactor giving 30-40% control of inlet HC, the second was a standard catalytic reactor giving 50-90% control depending on age, and the third was another swirl reactor giving 40-60% control. In turbocharged engines the number of reactor units was reduced to the second two, however some additional HC oxidation was probably obtained from the higher pressure and temperature levels existing between the engine ports and the exhaust turbine inlet. The amount of CO emission control required is relatively small, and does not present problems once catalytic reactors have been applied.

The primary emission target of 1.5 g/mile NO_x has been met by the M151 jeep without NO_x control techniques. However, other TCCS conversions generally require some combustion retard (i.e. the injection timing and ignition timing are retarded in phase), to achieve this level. The turbocharged Cricket in Fig. 2.15 was fitted with catalysts. In order to meet the secondary emission target, large quantities of exhaust gas circulation must be added to the engine. Typical EGR levels are 22% at light load and 11% at full load. To avoid compromising the engine volumetric efficiency, the gas is cooled in a heat exchanger to about 95°C. A number of TCCS vehicles in the 1130 and 1250 kg classes have met the secondary emission target as shown in Figs. 2.12 and 2.14 but, as already mentioned, with a considerable sacrifice in performance and fuel economy.

There are many results in the published literature referring to HC levels below 0.41 g/mile, with simultaneous NO_x levels in the region of 0.4 g/mile. However, there are no vehicle results (in the literature) for NO_x levels above 1 g/mile with a simultaneous HC level below 0.41 g/mile. In other words, the high EGR levels which result in slow inefficient combustion as noted above, raises the exhaust temperature and improves the catalyst efficiency. If EGR is not used, the engine is relatively efficient and exhaust temperatures are low due to the unthrottled state of the intake air. At low exhaust temperatures, the catalyst would not be operating over a portion of the CVS test. Therefore, an engine based on the TCCS process in its current development status may not be able to achieve the primary emission target. It may be necessary to use some intake air throttling to raise the exhaust temperature as in the Ford PROCO process. The optimum design and layout of injector and spark plugs would be influenced by this change in charge density. A corollary of this observation is that the TCCS process achieves secondary emission targets largely at the expense of fuel consumption, Fig. 2.24. The combustion speed could probably be optimised for high EGR levels, but there is a limit to this approach as hydrocarbon control would eventually become difficult.

A 50,000 mile durability test was conducted on a naturally aspirated L141 engine in an M151 jeep. However, the conditions were unrepresentative, as mileage was accumulated on a chassis dynamometer at a constant speed of 40 mile/h and this did not correspond to Federal procedures. A number of faults occurred in the emission control equipment during this test. First, the EGR system became partially blocked on two occasions and had to be cleaned. This would be a problem area on any TCCS engine, since smoke emissions are higher than from conventional gasoline engines. Second, each of the three reactors within the exhaust system had to be replaced once.

There does not appear to be a recognisable trend of vehicle hydrocarbon emissions as the fuel type is changed. In the turbocharged M151 mentioned above, CVS hydrocarbon emissions increased with heavier fuels, Fig. 2.14, but in the turbocharged Cricket, the opposite trend was observed, Fig. 2.15. The hydrocarbon level is incluenced by the method of measurement, i.e. whether the sample line was heated or unheated. The published literature on the M151 vehicle does not include this information, but in the case of the turbocharged Cricket the hydrocarbon level increased from 0.84 g/mile to 1.97 g/mile on diesel fuel when the measurement technique was changed. The first figure was obtained from the CVS cold bag technique, while the second involved integrating a continuous record. On gasoline fuel, hydrocarbon levels were not influenced by the measurement technique. Texaco state in their literature that HC levels from TCCS engines are independent of the fuel type, as heavier HC molecules from diesel fuel are broken down by the catalyst. This argument ignores the dominance of HC emissions during the cold start period when the catalysts are not functioning. In general, heavier fuels cause an increase in CO levels but no change in NO_x emissions.

Smoke.

The engines described under this category have a performance limit which is determined by the exhaust smoke level. For automotive applications, a smoke level of Bosch number 2½ is usually assumed to be permissible at low speed and 1½-2 at maximum speed. These figures correspond to air/fuel ratios in the region of 18 to 22 in the TCCS process. Performance curves for Curtiss-Wright engines indicate that maximum power occurs at an air/fuel ratio of 20 to 25, but there are no details on the smoke level used to fix this figure. Obviously, air utilisation in the Curtiss-Wright engines is restricted by the air trapped between the extremities of the rotor and the housing. This air cannot be mixed with the burning gases in the rotor cavity. Although smoke levels are included in performance curves from Deutz engines they have not been related to air/fuel ratio. However, air utilisation is probably comparable to the TCCS process.

Continuous recording of smoke levels have been taken from a turbocharged TCCS Cricket in the CVS-CH test, Fig. 2.28. On gasoline fuel, smoke levels are relatively low, but on diesel fuel the engine was apparently emitting unacceptably high smoke levels during accelerations. The minimum air/fuel ratio of this engine had been set higher than for naturally aspirated engines, i.e. 25 on gasoline and 23 on diesel and therefore the high smoke levels were caused by turbocharger lag as the fuelling was increased. The diesel engines used for comparison in Fig. 2.28 were naturally aspirated. Overall, the TCCS engine is comparable with a diesel engine in terms of smoke, but it is much inferior to other stratified charge engines. At 0.4 g/mile NO_x the high exhaust temperature which improved catalyst efficiency probably gave an added bonus in smoke control. Carbon particles can be oxidised on a platinum catalyst. However, if exhaust temperatures are low, carbon particles can be deposited on platinum catalysts and reduce the conversion efficiency. Smoke in the exhaust of TCCS engines, also leads to higher particulate emission levels than from homogeneous charge or other stratified charge engines, Figs. 5.21 and 5.22. However, the turbocharged TCCS Cricket engine in this particular build gave lower particulates than from two diesels.

Odour.

Composite odour intensity readings of the TCCS turbocharged Cricket obtained by a Turk panel are shown in Fig. 2.29. On gasoline fuel, the TCCS engine is slightly better than the average diesel engine, but on diesel fuel it is comparable with a poor diesel engine in this respect. As with smoke levels, the odour level of the TCCS engine is considerably higher than that of other stratified charge engines. In the diesel and TCCS engines, burnt components of the odour predominated.

Aldehyde emissions in the TCCS engine were high, Figs. 5.23 and 5.24, but ammonia was low, Figs. 5.25 and 5.26.

Topic Area (f) Multi-fuel Capability

The property of an engine to operate on a variety of fuels can be most important. The military application, where a combat vehicle may use any available hydrocarbon fuel, can have a major effect on combat logistics. In the future, if the powerplants of personal transportation could burn fuels without octane or cetane requirements, then the overall energy consumption of the world could be influenced. Texaco estimate, Ref. 2.1.7, that utilising a wide cut fuel in TCCS engines in 85% of the vehicle population could reduce U.S. national fuel consumption by 8-11%.

A multi-fuel engine is usually defined as one where one fuel can be substituted for another, without resetting operating parameters such as injection and ignition timing. This implies that the engine will be optimised for economy or emissions on one fuel, and that the operating conditions will not be optimised for all the fuels which that engine will burn.

Fuels.

TCCS engines have operated on a wide range of fuels, including the common gasolines, diesel and JP4 and 5. In addition, many tests have been conducted on a special "wide cut" fuel called 100-600 (indicating its boiling range). This fuel is a representative wide proportion of the hydrocarbons which can be easily refined from the crude oil, and had not been blended to any octane or cetane requirements.

It may be that the "true TCCS" process can be modified, to reduce the swirl requirements and increase performance. The cost of this modification is probably an engine octane requirement. Therefore, there may be a range of possible combustion chamber configurations similar to the TCCS process where performance and fuel consumption can be traded off against multi-fuel capability.

The multi-fuel tests carried out on the Deutz and Curtiss-Wright processes were less extensive than on TCCS. Deutz experimented with mixtures of gasoline and diesel fuel in various proportions, and showed that any mixture gave a practical fuel for the stratified charge engines. Curtiss-Wright state that their stratified charge engines have been developed for the less volatile JP4 and JP5 fuels, but some tests were carried out on gasoline and diesel, but at a loss in fuel consumption. Communications with Curtiss-Wright indicate that recent unpublished tests have corrected this fault.

Performance.

As already mentioned under Topic Area (c), the relative performance of TCCS and Deutz engines on gasoline and diesel fuels can be summarised as follows:

(a) At the same rack position, diesel fuel produces 10-20% more torque than gasoline.

(b) At the same smoke level, gasoline fuel produces 0-10% more torque than diesel fuel.

If a multi-fuel engine from this category were fitted to a vehicle, some feedback mechanism would be required to control torque output depending on the fuel being used. The ideal technological solution would be a fast responding smoke sensor on the exhaust pipe to control the maximum rack position. This would also be applicable to turbocharged engines, since transient smoke caused by turbocharger lag is a major problem area. There are simpler feedback control techniques, such as a fuel specific gravity measurement using floats.

Fuel Consumption.

While fuel continues to be marketed on a volume basis, diesel fuel will give 10-15% better economy than gasoline in TCCS and Deutz engines. However, there are some contradictions in the experimental data as to which fuel is best on a weight basis. The Deutz engine gave 3% better full load fuel consumption than on gasoline. On the L141 TCCS engine, there was no apparent difference between the fuels over the engine load range. These apparent contradictions may indicate different optimisation procedures (i.e. engine optimised on different fuels) in the tests.

Emissions.

Hydrocarbon emissions on heavier fuels are influenced by the measurement technique and the presence or absence of catalysts. Gasoline and diesel fuels may give similar hydrocarbon levels in the CVS test if sample lines are not heated, while hydrocarbons on diesel fuel may double when the sampling system temperature is kept above the dew point. Gasoline hydrocarbon emissions are not significantly affected by the sample temperature. There is often a relationship between carbon monoxide emissions and smoke level at full load. Feedback techniques to control maximum output as mentioned under performance would probably reduce CO emissions. Nitrogen oxide emissions do not appear to be affected by the fuel type.

Topic Area (g) Engine Components

Ignition.

A wide tolerance of a spark ignition engine to ignition timing is desirable, so that wear in drive trains and ignition equipment does not affect the ability of the engine to ignite the mixture. In this category of stratified charge engine, the tolerance is achieved by using long duration sparks. In an early Texaco paper it was stated that a spark duration of 25° allowed 10° latitude in ignition timing, although this may not apply to the present TCCS. In the Deutz system, a spark duration of 15° allowed 20° latitude in ignition timing, but this was reduced to less than 10° as the plug gaps eroded.

The high energy ignition system required to fire the mixture can cause plug erosion problems. Deutz experienced a three-fold increase in plug gap during a 300 hour durability test, and the spark gap could not be adjusted due to the plug design, Fig. 2.4. More exotic electrode materials have been tried, such as platinum/iridium,but there are manufacturing difficulties for mass production. The TTIS is apparently superior in this respect, as a single set of plugs lasted for 26,000 miles in a durability test. The rotary engine apparently has additional problems caused by deposits and requires anti-fouling plugs.

Injection.

A limitation of fuel injection equipment is that the required injection rate can only be achieved at one engine speed. Fluid and pipe compressibility causes pressure waves within the system and affect the opening and closing times of the injector. This appears to have been a problem in the development of TCCS, since air swirl and fuel injection could only be matched at one engine speed. As engine speed increased, the injection rate became slower, and contributed (with low volumetric efficiency) to poor high speed torque. TCCS engines have used distributor pumps rather than the in-line pump. Neither Deutz nor Curtiss-Wright literature mentions matching of the injection equipment over the speed range, and in the latter case this is surprising. One would expect the problems to become acute in an engine with a wide speed range such as the rotary type, There may be a case for further development of injection systems specifically for stratified charge engines, rather than relying on diesel engine technology. Pump injectors may be an attractive approach, as the injection process can be controlled more precisely.

High pressure injection apparently gives a distinct advantage with respect to injector durability, in comparison to low pressure types such as Ford PROCO. Spray characteristics and engine operation are not affected by carbonaceous deposits on the injector. A single set of injectors have operated in a TCCS engine for 50,000 miles without cleaning. Fuel injection pumps usually require lubrication by the fuel, and the lubricating properties of gasoline are virtually absent. Frequently, approximately 2% oil is added to the gasoline and this does not affect performance or emissions.

Other Components.

The Curtiss-Wright and Deutz engines required only the location of a spark plug or injector in an existing engine, although in the case of the Curtiss-Wright engine, modifications to the rotor cavity shape are required, to match the air and fuel flow characteristics. The TCCS L141 was adapted from a gasoline engine. The deep piston bowl increases the cylinder block height, and this was accommodated by a distance plate between cylinder block and cylinder head. In more recent engines, the cylinder block appears to have been modified for the new height and the plate was omitted.

Intuitively, the close arrangement of spark plug and injector in all the engines in this category would be expected to cause thermal stress problems in the metal bridge between the holes, particularly since the hottest burning gases are in this region. However, this does not seem to have happened, even in the supercharged Deutz engine where gas densities are high, or in the Curtiss-Wright rotary engine where combustion occurs twice as often.

The volumetric efficiency of the TCCS engines is compromised by the swirl ports or masked valves and inlet valve lifts have been increased by 15-20% in an attempt to improve volumetric efficiency. This imposes higher forces on the valves and camshafts and associated components, and wear rates can accelerate. In the 50,000 mile test on the L141 TCCS engine, the camshaft was replaced twice due to wear on the inlet cams. The materials of these components must be improved if high lift values are retained.

Topic Area (i) Mathematical Models

Computer modelling of stratified charge engines in this category is extremely complex, and this is reflected by the limited literature on the subject. The fuel burning process alone is difficult to describe accurately. In Category I engines the fuel was unevenly distributed throughout the combustion chamber, but it was in vapour form. Therefore, one of two single step combustion models could be assumed, i.e. lean air fuel mixtures burning to carbon dioxide, nitrogen, oxygen and water vapour, or rich mixtures burning to carbon dioxide, water vapour, nitrogen, oxygen, hydrogen and carbon monoxide. However, in the Category II engines, both liquid and vapour fuels are unevenly distributed as combustion commences, while more fuel is still being injected. There are a number of intermediate oxidation stages including carbon formation. In addition, liquid fuel is evaporating under the combined influence of gas velocity, gas temperature caused by partially oxidised neighbouring fuel, and radiant heat transfer from glowing carbon. Therefore, the gas properties during combustion cannot be predicted although properties before and after combustion are relatively easy to predict. For these reasons, no attempt has been made to model NO_x formation in this class of engine, where the molecular species are not known.

Bracco attempted to model the Curtiss-Wright rotary stratified charge engine, with the particular interest in establishing the spatial distribution of fuel within the combustion chamber, and locating the best spark plug position. The model took account of: (a) gas velocity at each position in the lobe and each crank angle position (b) injection rate (c) droplet size and distribution (d) rate of droplet evaporation (e) heat transfer (f) relative position of spark plug and injector. The author has apparently achieved the objective of identifying spatial distribution of fuel, but there are some gross simplifications within the model, which render it useless for performance or efficiency predictions; (a) a constant value of specific heat has been assumed, independent of the gas temperature or mixture strength (b) seal leakage has been ignored (c) combustion was assumed to occur at stoichiometric mixtures, so that extra fuel existing in overrich regions was assumed to remain in that unburnt stage until extra oxygen became available. Even with these gross simplifications, this complex model required a considerable amount of computer time to predict the air fuel mixture stratification at one engine condition.

Topic Area (j) Engine Size and Weight

Information has been published on the effect of the TCCS conversion of the L141 engine. Although the detailed design of the carburetted L141 does not follow current gasoline engine design practice, the overall engine dimensions and weights are representative of that size.

Table 2.1. DESIGN DATA FOR TCCS CONVERSION

	Carburetted	Naturally Aspirated TCCS	Turbocharged TCCS
Length	68.3 cm	69.8 cm	69.8 cm
Width	54.3 cm	62.2 cm	64.8 cm
Height	58.4 cm	59.2 cm	59.2 cm
Weight	150 kg	170 kg	176 kg (estimated)

There were two reasons for the increased width of the naturally aspirated version. The water pump was relocated to facilitate an injection pump drive and secondly, the distributor was mounted horizontally from the injection pump and at right angles to the engine centreline. Therefore, it protruded at one side in the overall engine silhouette. There was a further increase in engine width when the turbocharger was fitted. The cylinder block spacer, used to accommodate a taller piston, increased the engine height by 1.5 cm. Of the 20kg penalty of the TCCS conversion, 6kg can be attributed to the injection equipment. The remainder would arise from the injection equipment drive and associated engine modifications, the cylinder block spacer plate, and the pistons. These comparisons are based on identical engine sizes, and could therefore be misleading. A 2.3 litre (141 in^3) European engine would typically produce 63-67 kW (85-90 bhp). The naturally aspirated TCCS produced 43 kW (57 bhp) and the turbocharged Cricket TCCS 57 kW (77 bhp).

There are no details on engine size and weight for the Deutz engine, but the likely increase in weight over the original diesel would be about 1-3 kg due to the ignition equipment.

The rotary stratified charge engine has an advantage in size and weight over both conventional and stratified charge reciprocating engines.

Table 2.2. COMPARISON OF ROTARY AND RECIPROCATING ENGINES

	Carburetted Reciprocating Engine V8	Carburetted Rotary Engine RC2-60-U5	Stratified Charge Rotary Engine RC2-60-U10
Size	5.75 l	5.9 l	5.9 l
Power	135 kW	138 kW	119 kW
Length	76 cm	45.7 cm	60.8 cm
Width	63 cm	56.0 cm	60.8 cm
Height	74 cm	54.8 cm	60.8 cm
Weight	270 kg	107 kg	133 kg

Since these figures, published by Curtiss-Wright, seemed remarkably favourable towards the rotary engine, the weights of reciprocating and rotary automotive engines are compared in Fig. 2.30. As shown by this graph, the Curtiss-Wright engines are considerably lighter than most rotary engines. The original Curtiss-Wright rotary engines from which the stratified charge engine was derived was intended for aero applications where weight is at a premium, and more costly manufacturing techniques are justified for weight reduction.

Topic Area (k) Noise and Vibration

Cylinder pressure diagrams for engines in this category are remarkably smooth, and rates of pressure rise are usually in the region of 2½ bar/degree (30 psi/degree) as for a conventional gasoline engine. In addition, the diagrams remain relatively stable as load is reduced, and this can give a noise improvement over the gasoline engine, which suffers from cyclic irregularity. Maximum cylinder pressures also influence the noise level. The maximum pressure at full load in the naturally aspirated TCCS engine is lower than the gasoline engine, 45-48 bar (650-700 lbf/in^2) compared to 55-62 bar (800-900 lbf/in^2) due to the low specific torque output. However, as load is reduced, cylinder pressure in the unthrottled engine becomes higher than in a gasoline engine. Therefore, idle noise would be expected to be greater than from the gasoline engine.

Noise measurements carried out at SWRI on the turbocharged TCCS Cricket and other stratified charge engines are shown in Figs. 2.31-2.33. The full load maximum speed test SAE J986a cannot be interpreted at a glance, as there are a number of important factors governing noise, i.e. engine bore size, engine speed and type of engine structure. An empirical Ricardo relationship between these factors has been derived and correlated with anechoic cell measurements, and this relationship was used to derive predicted noise levels, Fig. 2.33. The absolute noise level of the TCCS engine is low in this test due to the restricted speed range. As load is reduced, the TCCS engine becomes relatively more noisy than throttled engines, and does not differ from the diesel in this respect. The TCCS engine is partially throttled which should reduce the cylinder pressures compared with the diesel. Changing the fuels from gasoline to diesel only increased noise levels at idle.

There are no results on noise levels from rotary engines, although in-house experience suggests rotary engine levels could be reduced below reciprocating counterparts, assuming that pressure diagrams were similar. The carburetted

Mazda RX2 rotary engine had similar noise levels to reciprocating engines, but this was shown to arise from the induction process, and combustion radiated noise was 0-5 dBA lower (depending on position of microphone) than a comparable reciprocating engine at the peak power point. The two-thirds epitrochoid structure is inherently more rigid than a cylinder block. It was concluded that better intake design could significantly improve the rotary engine noise levels. The same reasoning would probably apply to a comparison of unthrottled rotary and reciprocating engines in this category.

Vehicle vibration is excited by the instantaneous torque fluctuations between the engine and drive train, and this is usually most important at low engine speeds and idle. Cylinder pressure diagrams from diesel and gasoline engines were used to calculate the instantaneous indicated crankshaft torque for a 6 cylinder engine, 95 mm bore x 95 stroke. Torques generated by the gas pressure and inertia forces from the reciprocating components were both considered. As shown in Fig. 2.34 the gasoline engine has a relatively smooth torque profile compared to the diesel, due to the throttled intake and lower compression ratio. The torque reversals are caused by the compression stroke. The same exercise was repeated using cylinder pressure diagrams from a stratified charge engine with a compression ratio of 12. Diagrams were available for completely unthrottled idle and for partial throttling. When unthrottled, the level or torque fluctuations lay mid-way between the diesel and gasoline engines, but even a small amount of throttling significantly reduced the torque fluctuations. Therefore there is a good case for throttling stratified charge engines at idle and low loads for (a) lower noise (b) less vibration (c) avoiding excessive hydrocarbons.

Topic Area (1) Durability and Starting

Durability.

The only engine mechanical durability problem mentioned in the references concerns the valve gear of the L141 TCCS engine. Since valve lifts are higher than usual (i.e. up to 35% of the intake valve diameter) the forces imposed on the tappets are high. In the 50,000 durability test on the M151 jeep, the valve gear had to be replaced twice. The problem was overcome using better tappet materials. Deutz stratified charge engines have operated in trucks on the highway, and the only unusual servicing compared with the diesel, is the replacement of spark plugs at 15,000 km (10,000 mile) intervals. There do not appear to have been any durability tests carried out on the Curtiss-Wright stratified charge engines. Seal tip wear which plagued early rotary engines appears to have been solved, and stratified charge combustion should not impose any greater loads on these components.

Cold Starting.

Table 2.3 shows figures quoted for cold start tests:

Table 2.3. COLD-START TEST RESULTS

	Temperature	Fuel	Comments
Curtiss-Wright RC2-60-U10	-37°C	JP4	
TCCS L141 Naturally Aspirated	-32°C	Gasoline CITE Winter grade diesel	
TCCS L144 Naturally Aspirated	-40°C -46°C	Gasoline Gasoline	Battery warmed and crankcase diluted
Deutz Stratified Charge	-15°C	Gasoline Diesel	Would also start on compression ignition
	-30°C	Gasoline	Torch heater system for intake air

The torch heater on the Deutz engine consisted of a separate pump which was driven electromagnetically. This pump supplied fuel to the intake manifold which was then ignited by a plug. The system was arranged to be independent of the type of fuel, i.e. diesel or gasoline. The reason for the poorer starting characteristic of the Deutz engine is not known.

Application to Motorcycles

The loss in specific power output would be even more spectacular than Category I combustion systems, and the cost of the fuel injection system would be equally unattractive. The high HC levels emitted by these engines at low loads might cause the motorcycle target HC emission levels to be exceeded. However, the target levels could probably be achieved with development, and so avoid the use of a catalytic reactor.

The TCCS process has been applied to a small engine of two cylinders, 35 cc swept volume, and the specific power output was 15.2 kW/litre.

Conclusions

Since the conclusions in this category are complicated by emission levels, types of fuel, turbocharging, etc., a guide is given in Table 2.4. The figures quoted are approximations based on the literature survey.

1. Three principal processes have been identified in this category, TCCS and Deutz which are based on the reciprocating engine configuration and Curtiss-Wright, which is based on the rotary engine.

2. The description of TCCS as a combustion system which is controlled by the rate of fuel injection is over-simplified. Combustion in all the systems in this category is extremely complex and depends on many other parameters besides the fuel injection rate.

3. Maximum engine torque is always limited by smoke emissions, and air utilisation in these engines is relatively poor. Specific torque (i.e. BMEP) is below that of the indirect injection diesel engine and far below that of the gasoline engine. In addition, the high air swirl requirements compromise the high speed volumetric efficiency, so that the engine speed

Table 2.4. GUIDE TO ENGINE CHARACTERISTICS

		DIESEL ENGINE	GASOLINE ENGINE	CATAGORY 2 STRATIFIED CHARGE ENGINE				
				RECIPROCATING				ROTARY NATURALLY ASPIRATED GASOLINE FUEL
				TURBOCHARGED		NATURALLY ASPIRATED		
				GASOLINE FUEL	DIESEL FUEL	GASOLINE FUEL	DIESEL FUEL	
ENGINE SWEPT VOLUME (FOR SAME POWER OUTPUT)		125%	100%	108%	117%	117%	152%	140% (5) AS 3 LOBES)
FUEL ECONOMY					SEE CONCLUSION(7)		SEE CONCLUSION(7)	SEE CONCLUSION(3)
UNCONTROLLED EMISSIONS		[illegible]	100%	108%	117%	100%	111%	CONCLUSIONS SAME AS RECIPROCATING
PRIMARY TARGET		[illegible]	97%	N.P. SEE CONCLUSION(5)	NP	NP	NP	
SECONDARY TARGET		NOT POSSIBLE	90%	72%	80%	70%	78%	
PERFORMANCE LOSS AT SECONDARY TARGET		NOT POSSIBLE	< 5%	≥ 30%	≥ 30%	≥ 30%	≥ 30%	
NO OF CATALYSTS OR THERMAL REACTORS AT	PRIMARY TARGET	0	1	NP SEE CONCLUSION(5)	NP	NP	NP	BASELINE HC LEVELS VERY HIGH, MAY NOT BE POSSIBLE TO REACH 0.4 GM/MILE HC
	SECONDARY TARGET	NA	2	2	2 (BASELINE HC LEVEL HIGHER THAN GASOLINE)	1	3 (BASELINE HC LEVEL HIGHER THAN GASOLINE)	
ODOUR		MODERATE	LOW	MODERATE	HIGH	MODERATE	HIGH	CONCLUSIONS SAME AS RECIPROCATING
PARTICULATES		MODERATE	LOW	PROBABLY SAME AS DIESEL ENGINE				
SMOKE		MODERATE	LOW	HIGHER THAN NATURALLY ASPIRATED DUE TO TURBO LAG		MODERATE	MODERATE	
COLD STARTING		-25°C (HEATER PLUGS)	DOWN TO -40°C	-12°C	-12°C	-40°C	-40°C	
NOISE	FULL LOAD FULL SPEED	MODERATE	MODERATE	LOWER THAN GASOLINE DUE TO SMALL BORE AND LIMITED SPEED				
	IDLE	HIGH	LOW	MODERATE	HIGH	MODERATE	HIGH	
ENGINE WEIGHT (FOR SAME POWER)		[illegible]	100%	105%	117%	120%	125%	[illegible]

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range and maximum power are also low. The rotary engine does not suffer from this speed limitation, and there is probably room for further development to higher maximum speeds, if suitable injection equipment becomes available. However, the problems of seal leakage and seal life in rotary engines must be solved, if the lower fuel consumption potential of stratified charge is to be realised in these engines.

4. Test bed fuel consumption on some TCCS engines has been exceptionally good, surpassing both the gasoline and diesel engines at low load. However, this has not translated to vehicle applications, where fuel economy has only been marginally better than good gasoline engines. This may have been caused by the transmission characteristics of the vehicles chosen for conversion to TCCS or re-calibration of the engine for low emissions. The torque curve of TCCS engines can be easily matched to a turbocharger, but since the turbocharger does not materially affect low engine speed power output, an automatic transmission is desirable to raise the engine operating speed. The automatic transmission involves some loss in fuel economy.

   At the secondary emission target, fuel economy of TCCS engines is worse than that of the gasoline engine when it is tuned for dual catalytic reactors.

5. Baseline hydrocarbon levels from all the engines in this category are high, about 2.5-3.5 g/mile for TCCS, and greater, from the rotary engine.

   There are no published results on TCCS engines with HC levels of 0.41 g/mile and simultaneous NO_x levels around 1.5 g/mile, i.e. the primary emission target. It is suspected that the low exhaust temperatures inhibit the effectiveness of catalytic reactors, and the required degree of HC control cannot be achieved. This is probably a fundamental limitation of unthrottled stratified charge engines, and can only be circumvented by two methods (a) a 50-70% reduction in engine baseline HC levels or (b) throttling the engine to near stoichiometry to raise the exhaust temperature, but with the associated loss in fuel economy.

6. TCCS engines have met the secondary emission target by using large quantities of EGR to the point where the engine becomes inefficient, and the resulting high exhaust temperature aids the catalytic oxidation of the unburnt hydrocarbons. Considerable catalytic control is required to achieve 0.41 g/mile HC.

7. The reciprocating engines in this category have an excellent multi-fuel capability. Heavier fuels, such as diesel, give better fuel economies than gasoline, due to their greater specific gravities. If fuel were marketed on the basis of weight rather than volume, there would not be a difference in fuel economy between the fuels.

   There is very little information on the multi-fuel capability of the rotary version of this combustion process.

   If a multi-fuel engine were to be marketed in a vehicle, some method of controlling torque output would need to be incorporated in the fuel injection pump, based on the fuel density or the exhaust smoke level.

There is a loss in smoke limited power with the heavier fuels, and hydrocarbon emissions increase. The latter may not be observed unless the sample lines are heated.

8. Most of the mechanical durability problems with TCCS engines, have, or can be solved. There are no insoluble manufacturing problems, but capacity for a high demand of diesel type fuel injection equipment does not exist. The durability of the emission control equipment has yet to be proven, as the previous test was not carried out to the Federal procedures. There are likely to be difficulties with the durability of the thermal and catalytic reactors, and clogging of the EGR system with carbon particles.

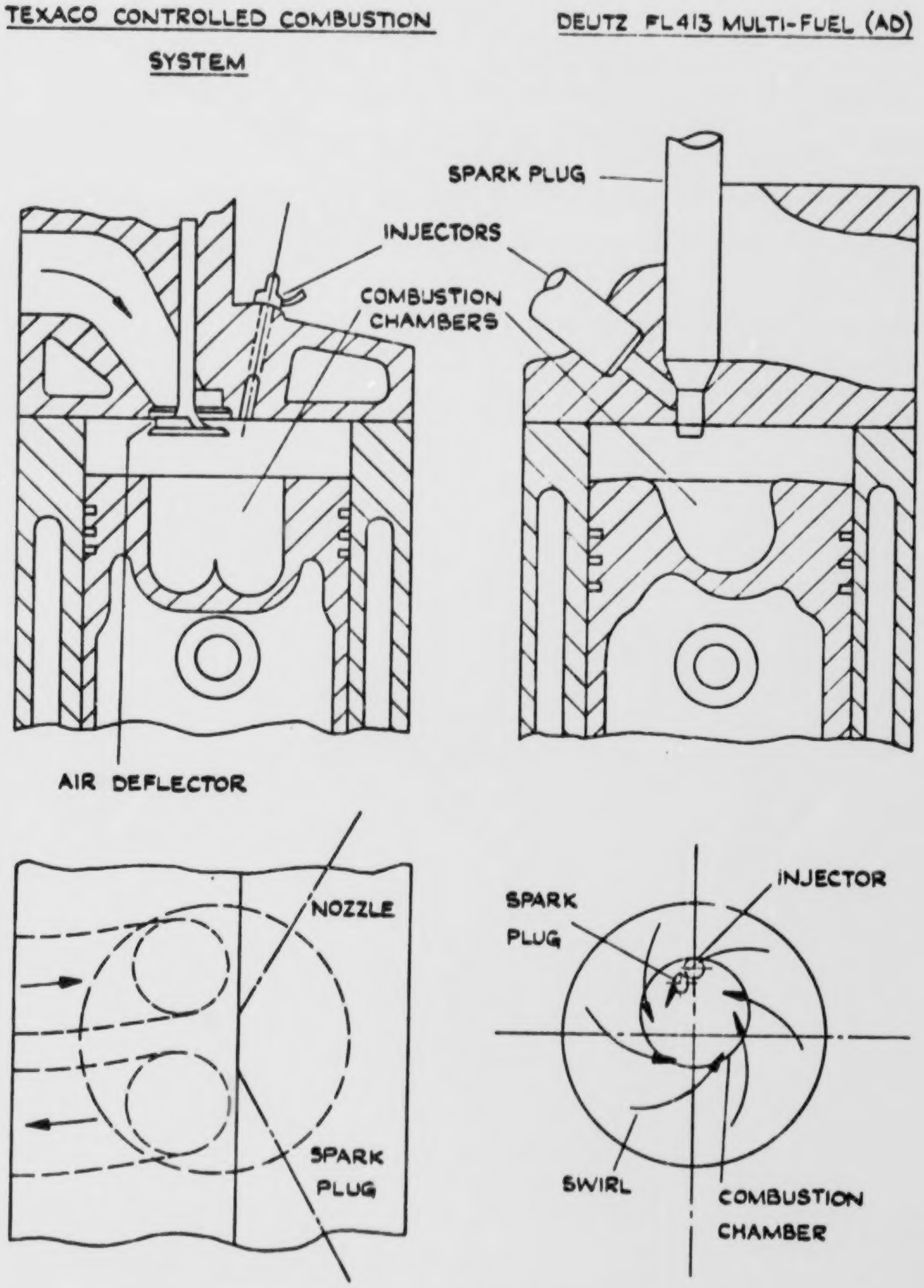
9. Additional problem areas identified with TCCS engines are:

   (a) High noise levels at idle.

   (b) Turbocharger lag during accelerations, causing high instantaneous smoke emissions.

   (c) Odour particularly when operating on diesel fuel.

Figure 2.1. Texaco and Deutz engines.

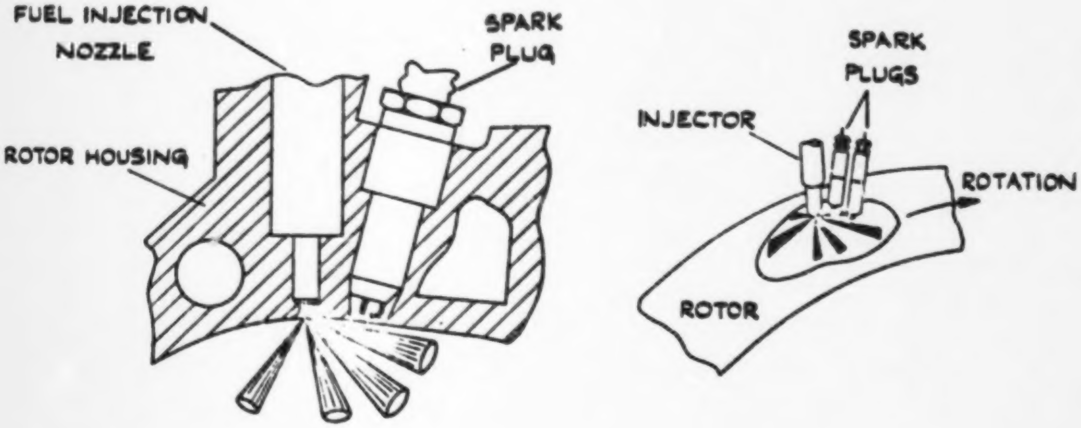


TEXACO CONTROLLED COMBUSTION SYSTEM

DEUTZ FL413 MULTI-FUEL (AD)

SPARK PLUG
INJECTORS
COMBUSTION CHAMBERS
AIR DEFLECTOR
NOZZLE
SPARK PLUG
SPARK PLUG
INJECTOR
SWIRL
COMBUSTION CHAMBER

Figure 2.2. Stratified charge RC engine showerhead nozzle injection configuration.


FUEL INJECTION NOZZLE
SPARK PLUG
ROTOR HOUSING
SPARK PLUGS
INJECTOR
ROTATION
ROTOR

STRATIFIED CHARGE INJECTION CONFIGURATION

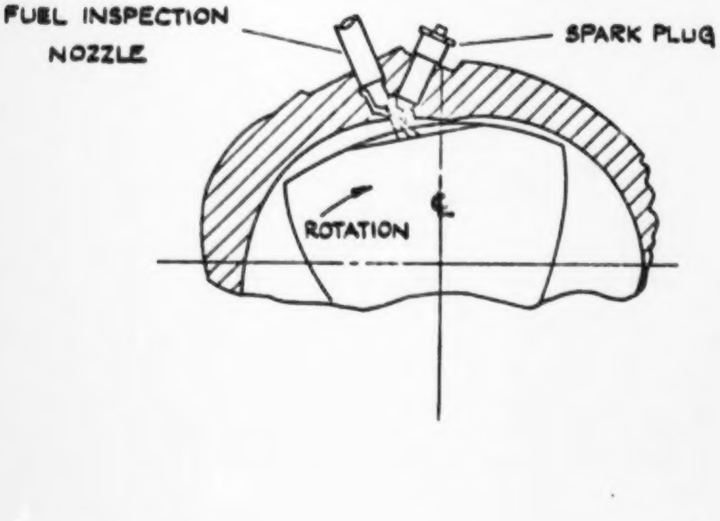

FUEL INSPECTION NOZZLE
SPARK PLUG
ROTATION
℄

Figure 2.3. Nozzle assembly.

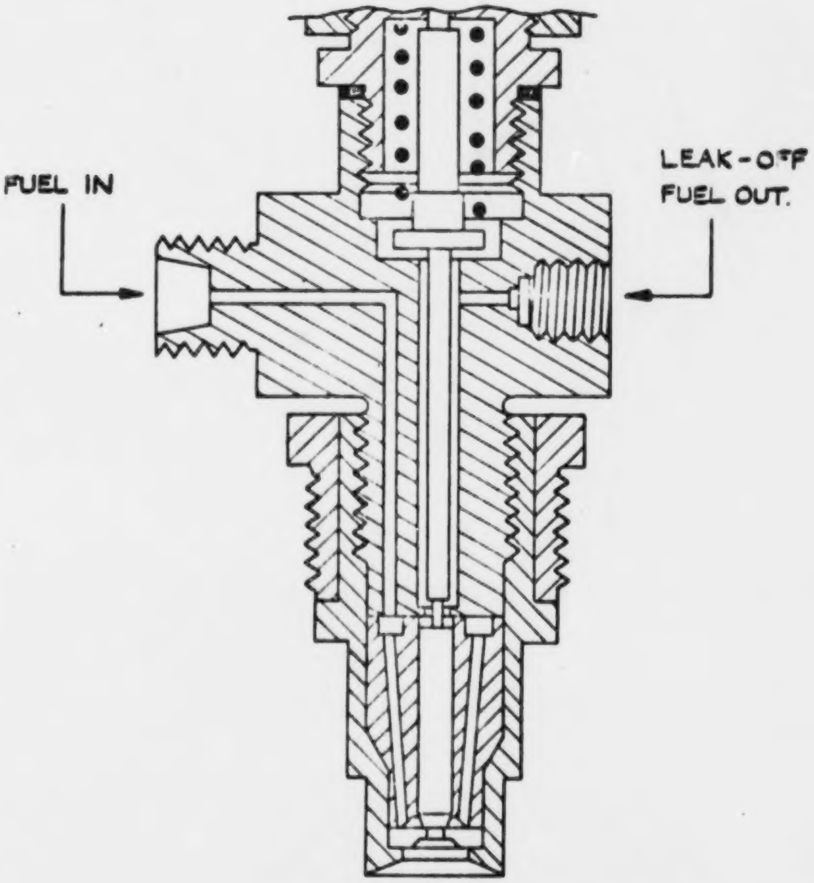
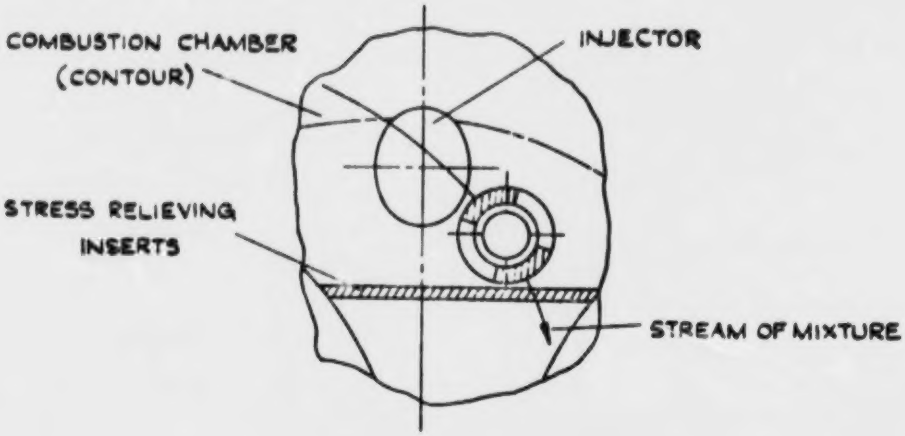

FUEL IN
LEAK-OFF
FUEL OUT.

Figure 2.4. Deutz FL413 multi-fuel (AD) injector, mixture flow and sectional view of spark plug.


COMBUSTION CHAMBER (CONTOUR)
INJECTOR
STRESS RELIEVING INSERTS
STREAM OF MIXTURE

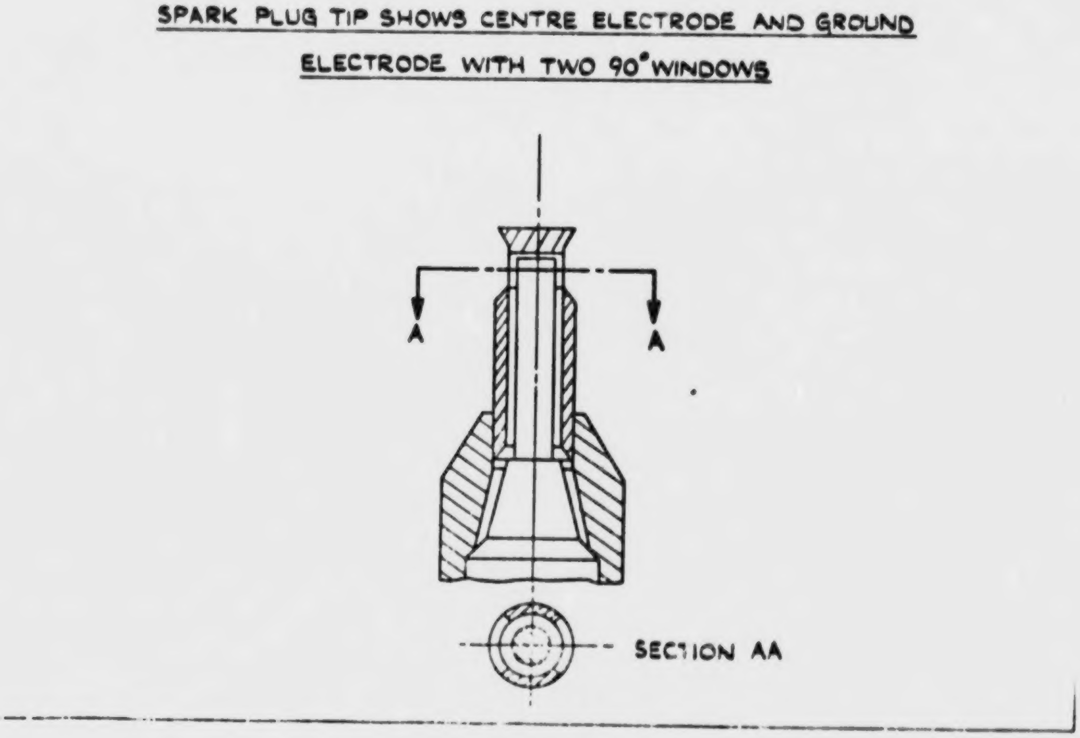

SPARK PLUG TIP SHOWS CENTRE ELECTRODE AND GROUND ELECTRODE WITH TWO 90° WINDOWS
A
A
SECTION AA

Figure 2.5. Stratified charge RC engine dual showerhead nozzle injection configuration.

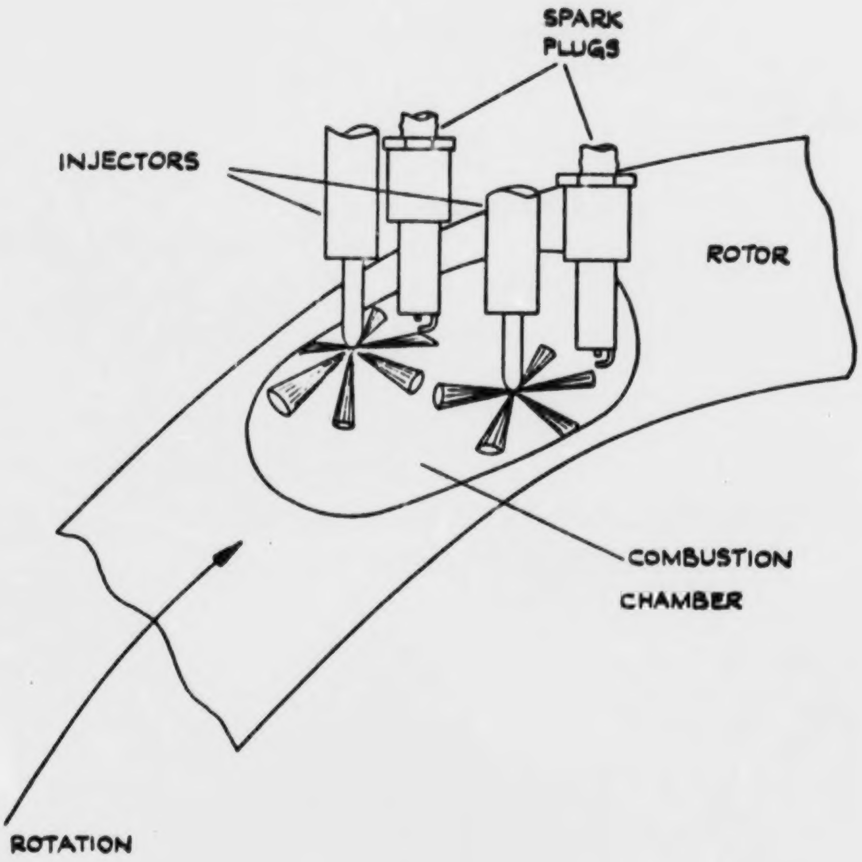

SPARK
PLUGS
INJECTORS
ROTOR
COMBUSTION
CHAMBER
ROTATION

Figure 2.6. Fuel injected - mass burned comparison of TCCS engine.

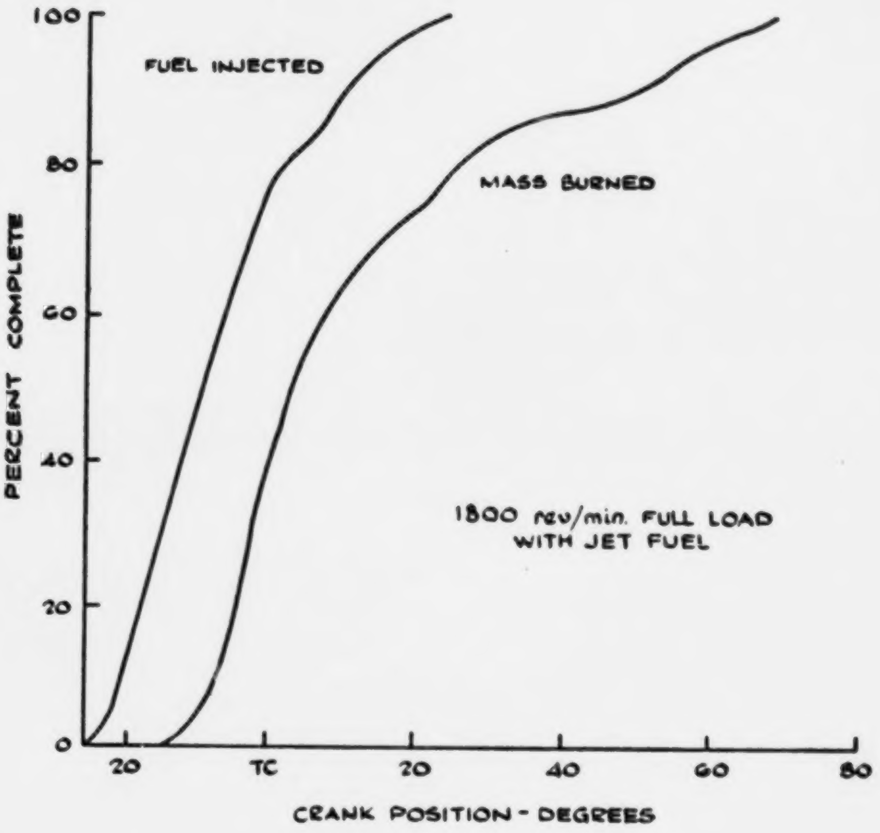
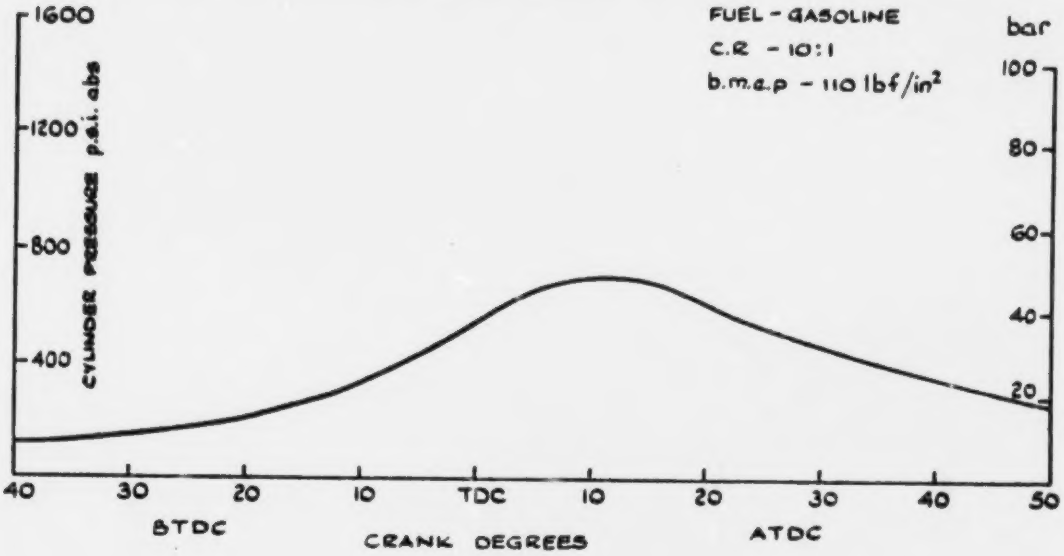


Figure 2.7. Comparison of cylinder pressure diagrams between a TCCS and a supercharged Deutz engine.

CYLINDER PRESSURE DIAGRAM FOR TCCS ENGINE AT FULL LOAD



CYLINDER PRESSURE DIAGRAM FOR SUPERCHARGED DEUTZ ENGINE

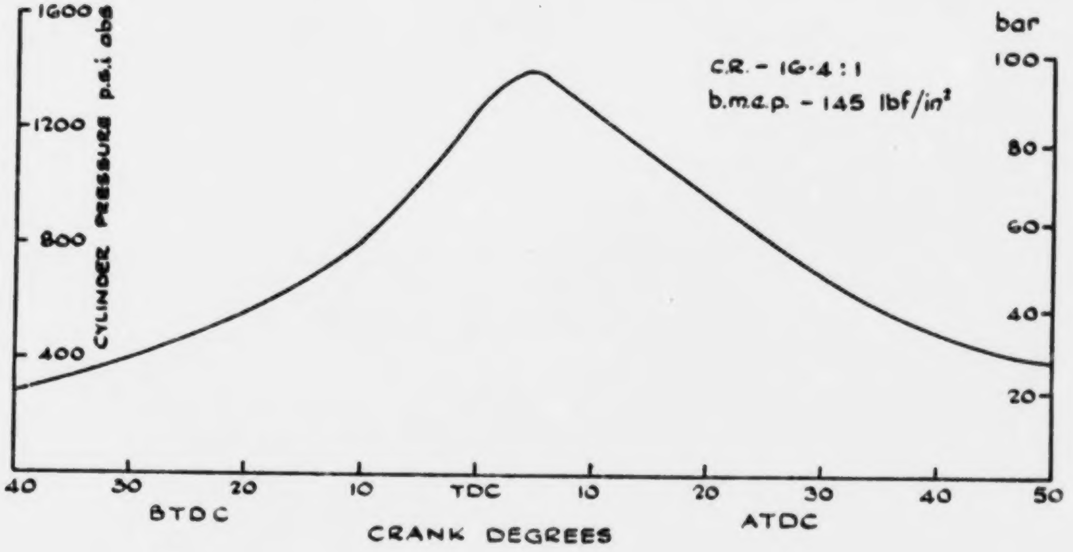


Figure 2.8. Comparison of torque curves of diesel and Curtiss Wright stratified charge rotary engine interpolated to 96 kw (data from modified results of RC2-60U10).

DIESEL ENGINE SWEPT VOL 4·77 l. (291 in³) MAX POWER SPEED 4000 rev/min.
ROTARY ENGINE SWEPT VOL 5·44 l. (338 in³) MAX POWER SPEED 5000 rev/min.
SWEPT VOL = TOTAL OF ALL LOBES OF ROTARY

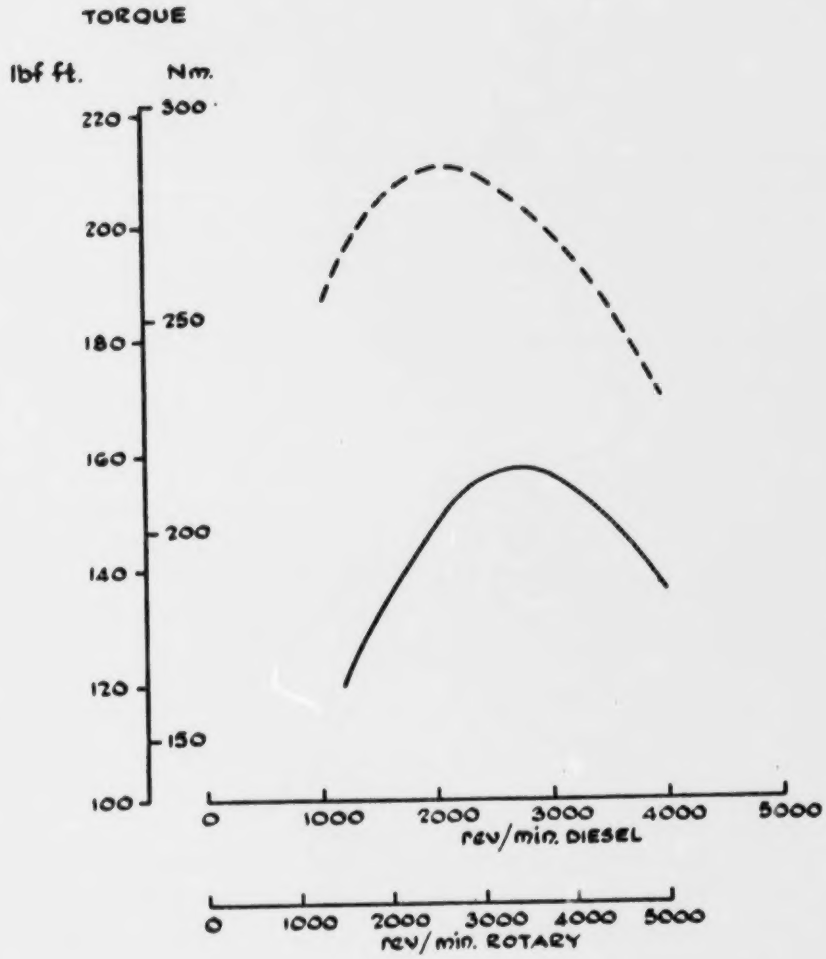


Figure 2.9. Specific output of TCCS engines.

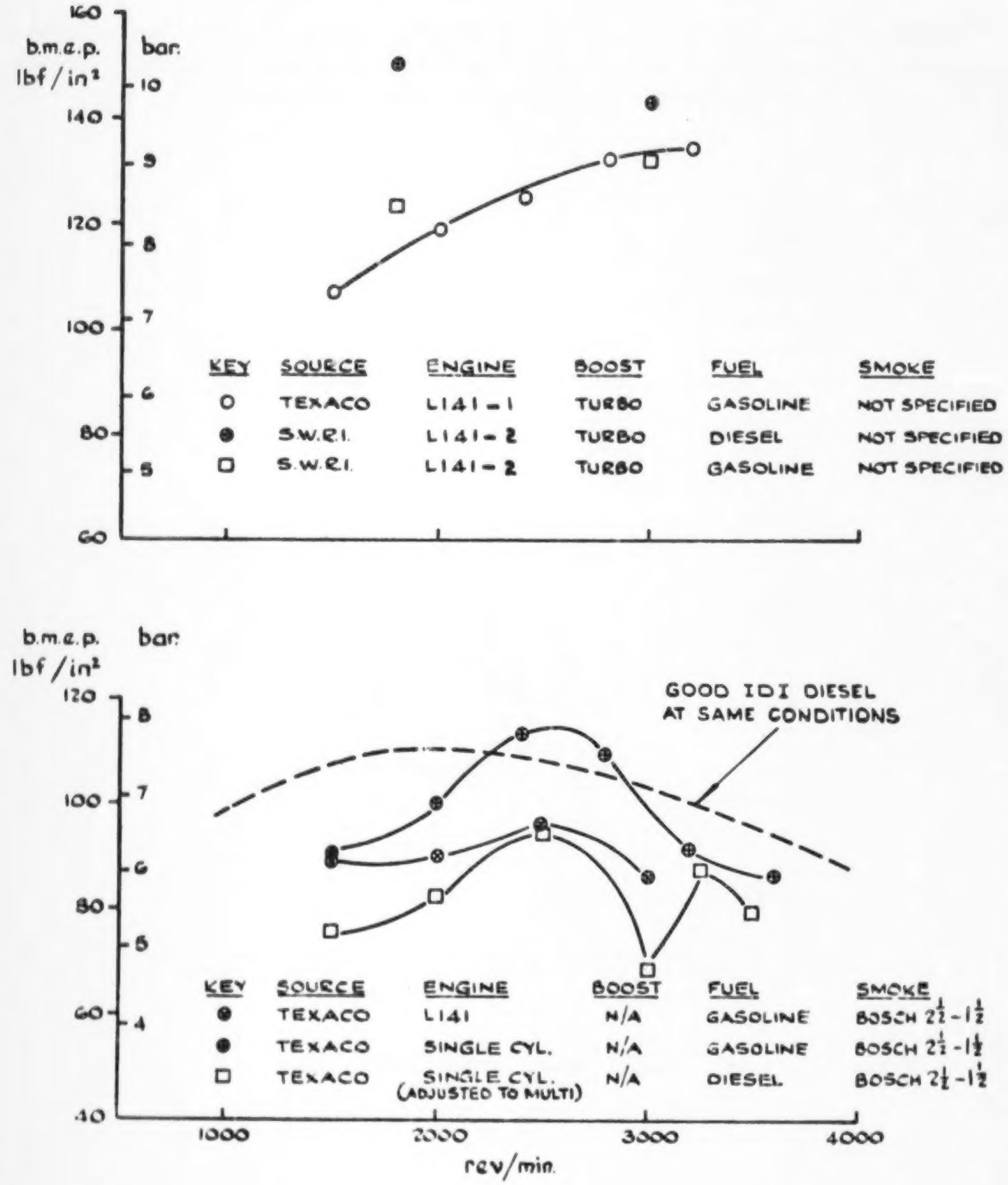


Figure 2.10. Comparison of smoke limited b.m.e.p. of gasoline, cetron and diesel on a LIS-188 engine (at early stage of development).

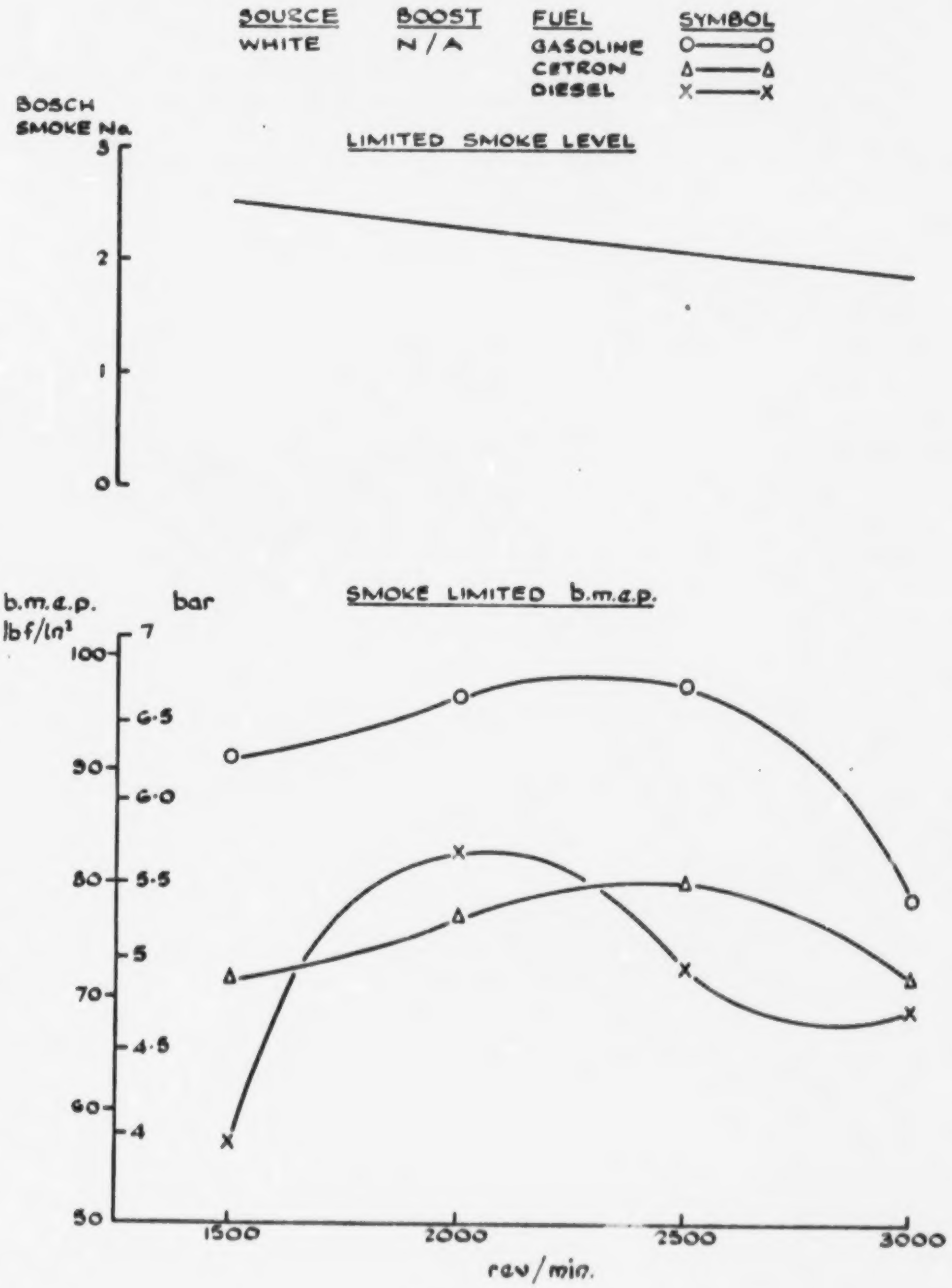


Figure 2.11. Specific output of Deutz engines turbocharged and intercooled Bosch smoke No. 2½ - 1½.

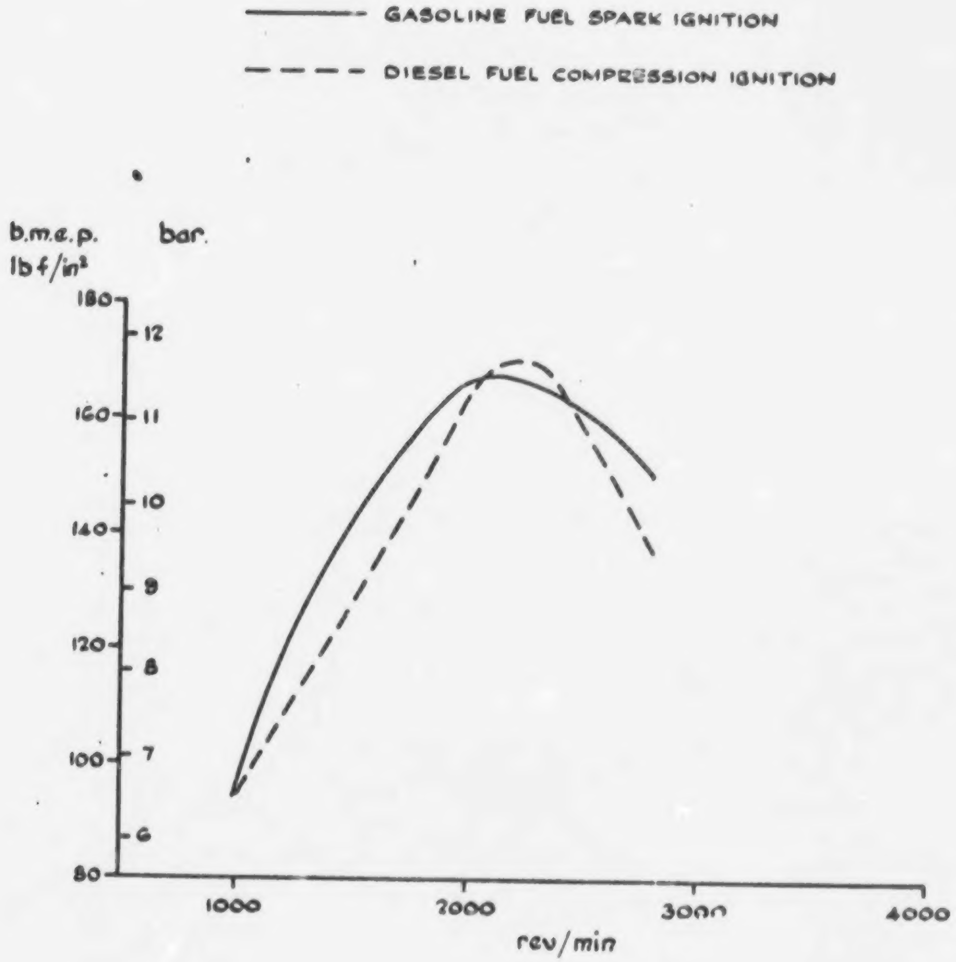


Figure 2.12. Emission results from catalytically controlled TCCS vehicles.

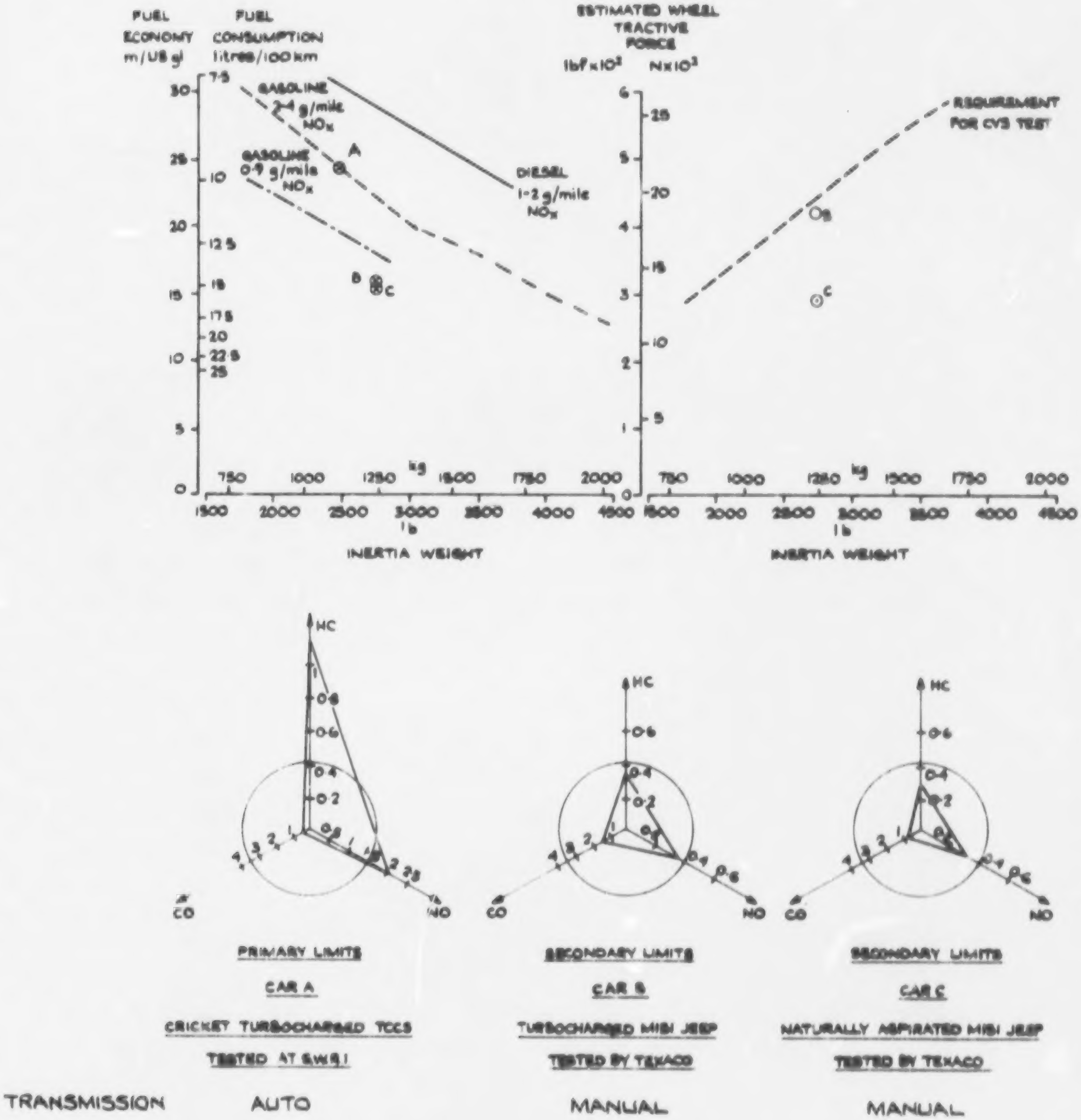


Figure 2.13. Emission results from TCCS vehicles without emission controls.

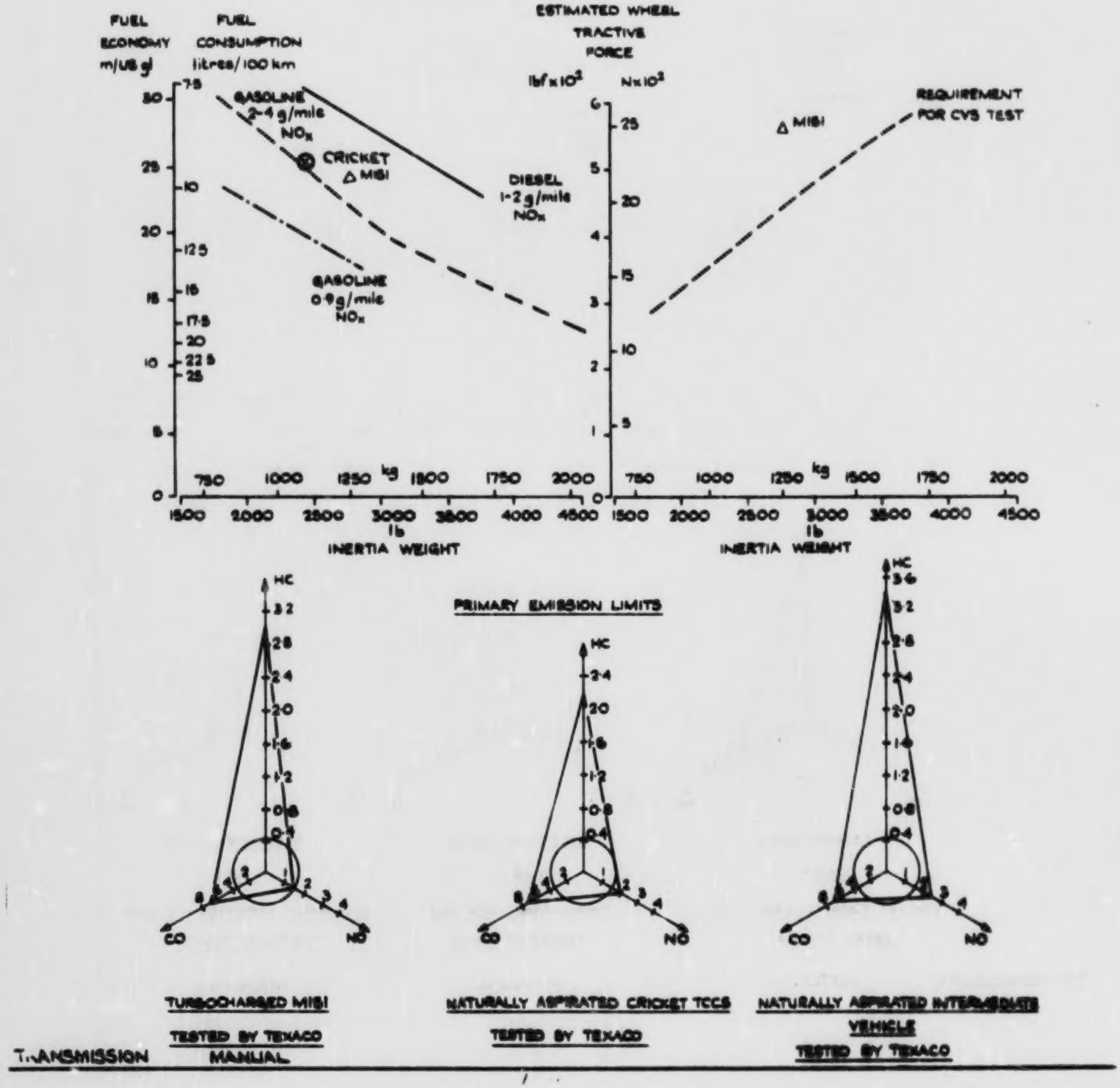


Figure 2.14. Effect of fuel type on emissions and economy of TCCS M-151 vehicle results after 50,000 miles durability, high EGR rate, 3 catalysts.

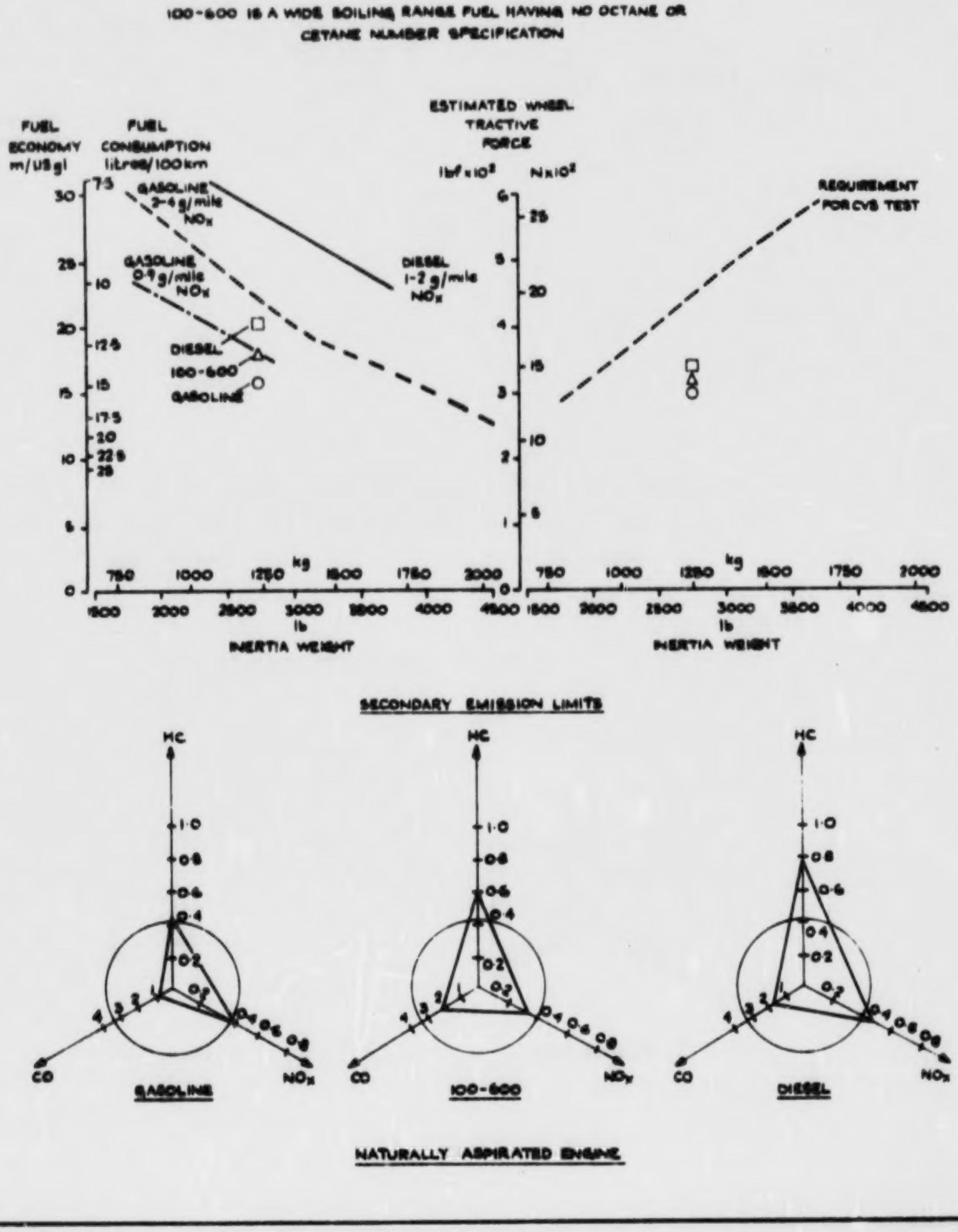


Figure 2.15. Effect of fuel type on emissions and economy of TCCS Cricket (SWRI results).

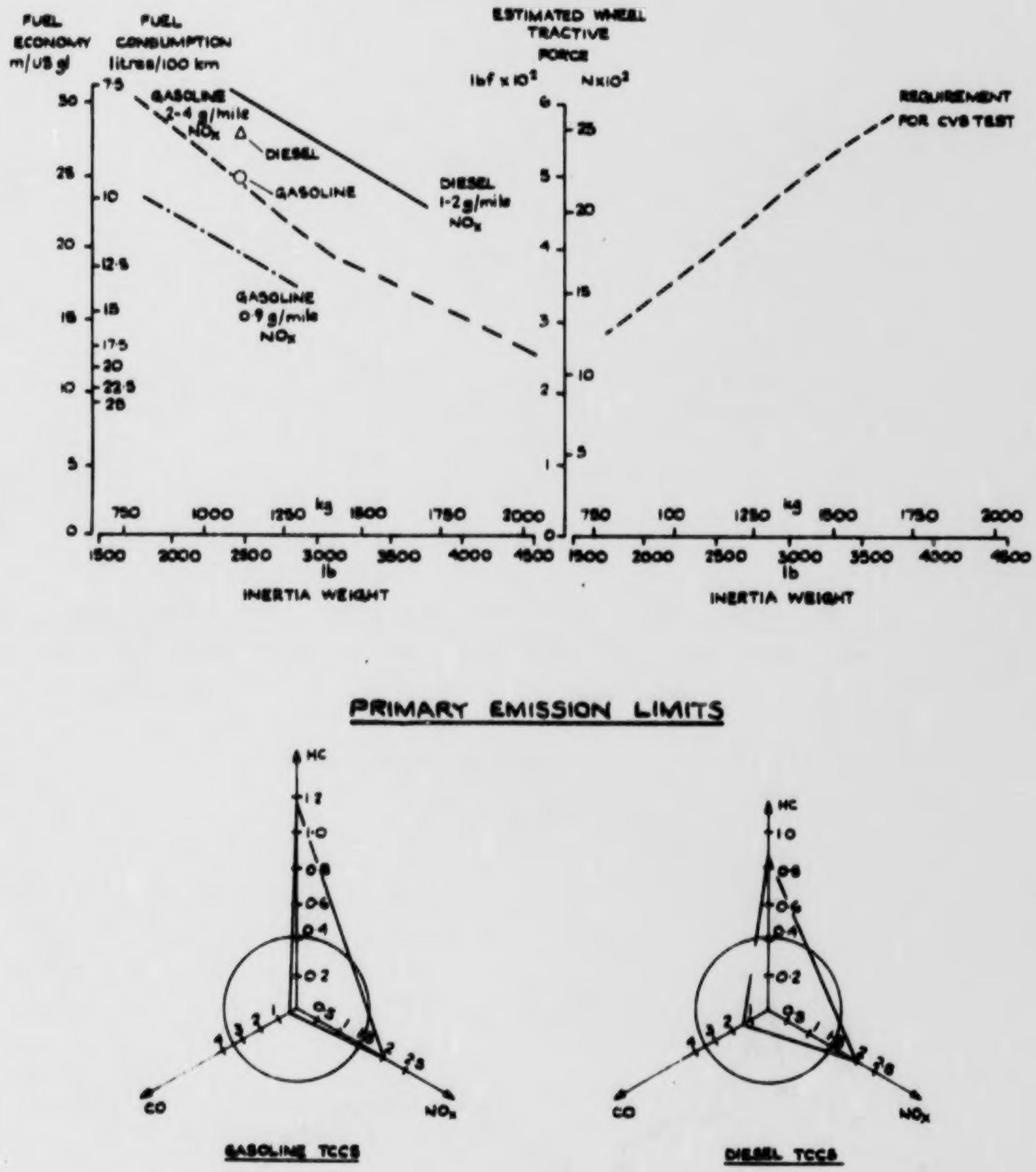


Figure 2.16. Motoring friction of L141 TCCS engines.

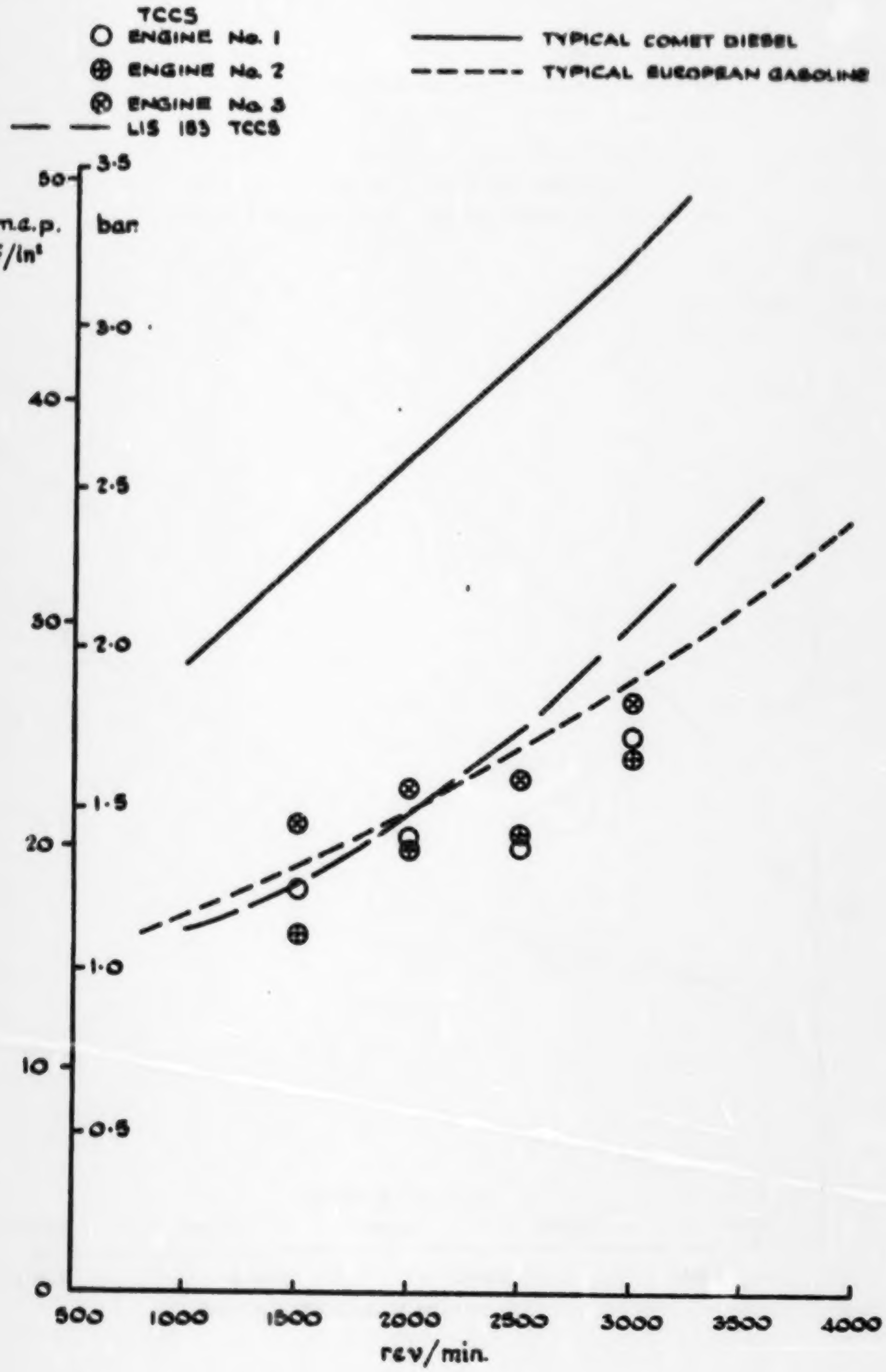


Figure 2.17 Comparison of friction levels of rotary and reciprocating engines.

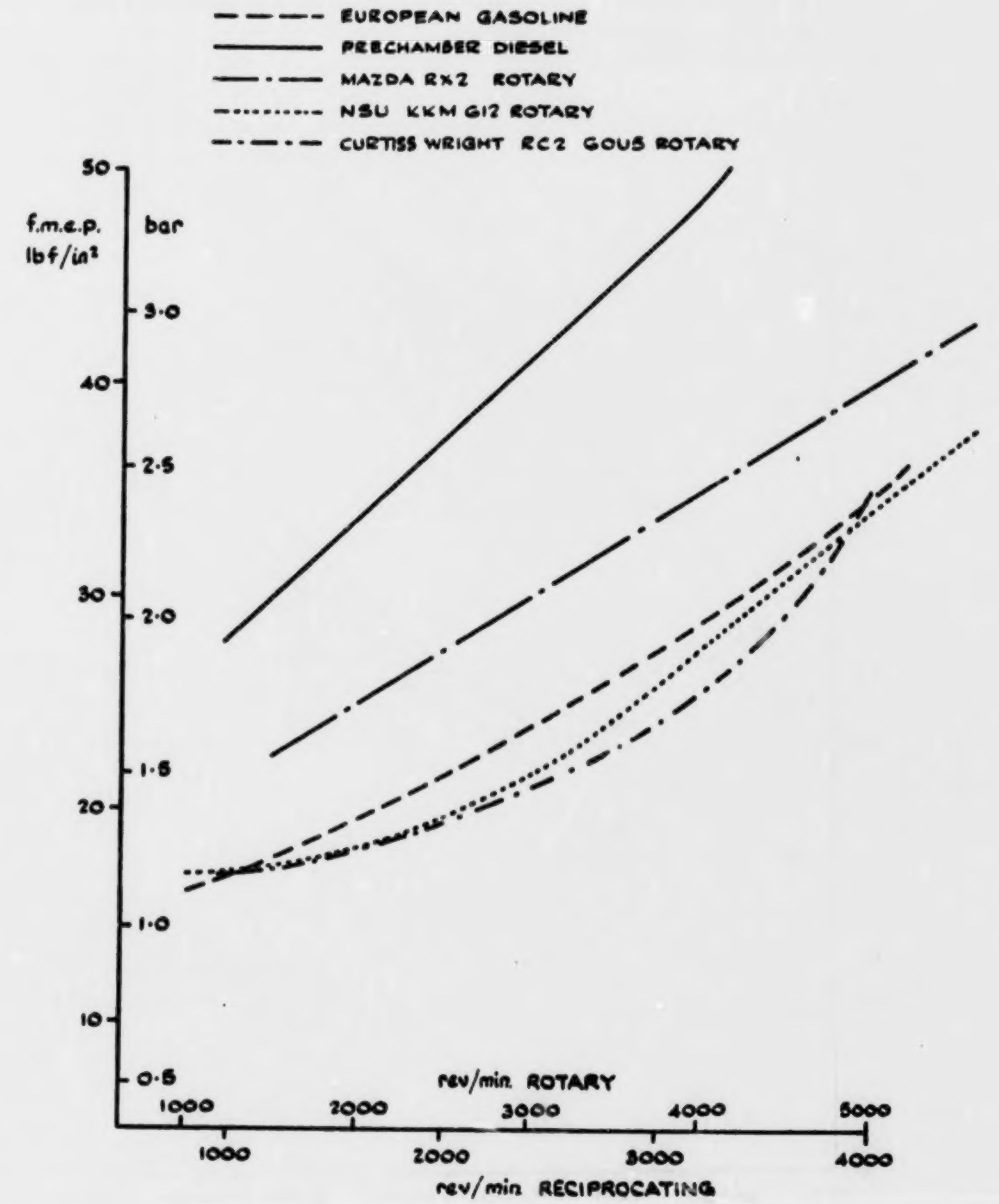


Figure 2.18. Break-down of motoring friction of Mazda Rx2 and 2 litre reciprocating engine.

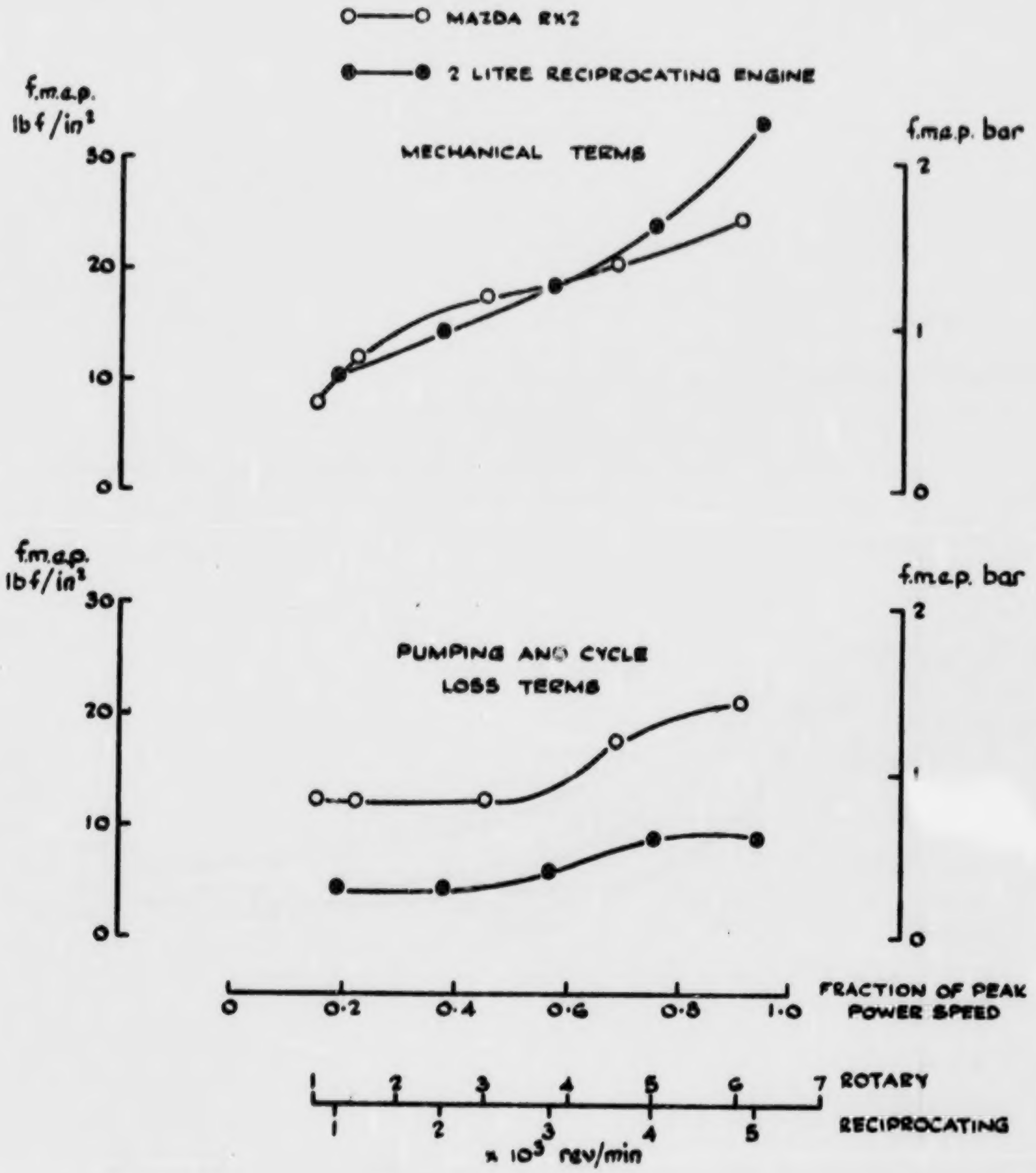


Figure 2.19. Comparison of fuel consumption between and RCI - GO stratified charge engine (C.R. 8.5:1) interpolated to target horsepower and baseline diesel engine.

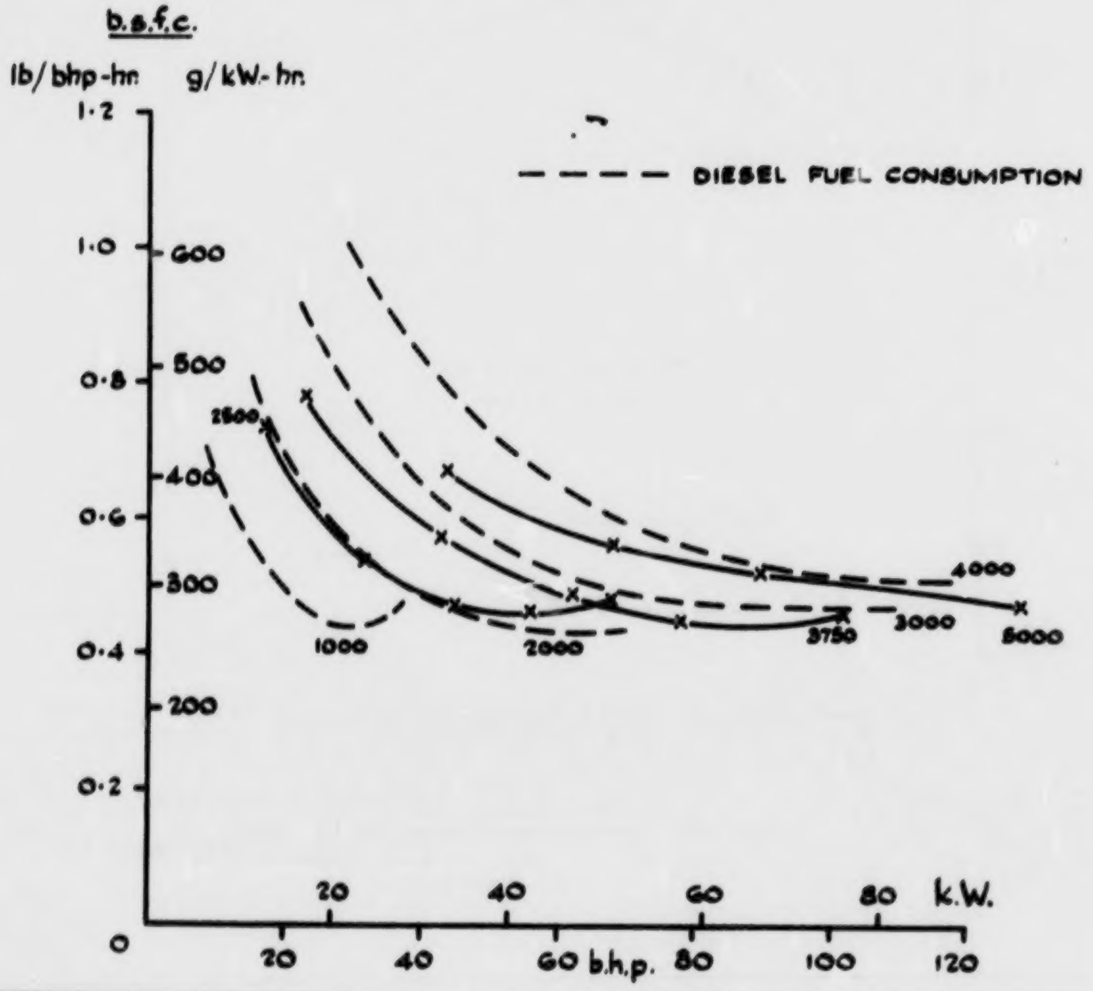


Figure 2.20. Comparison of emissions and fuel consumption of a diesel engine and a RCI - 90 stratified charge rotating combustion engine.

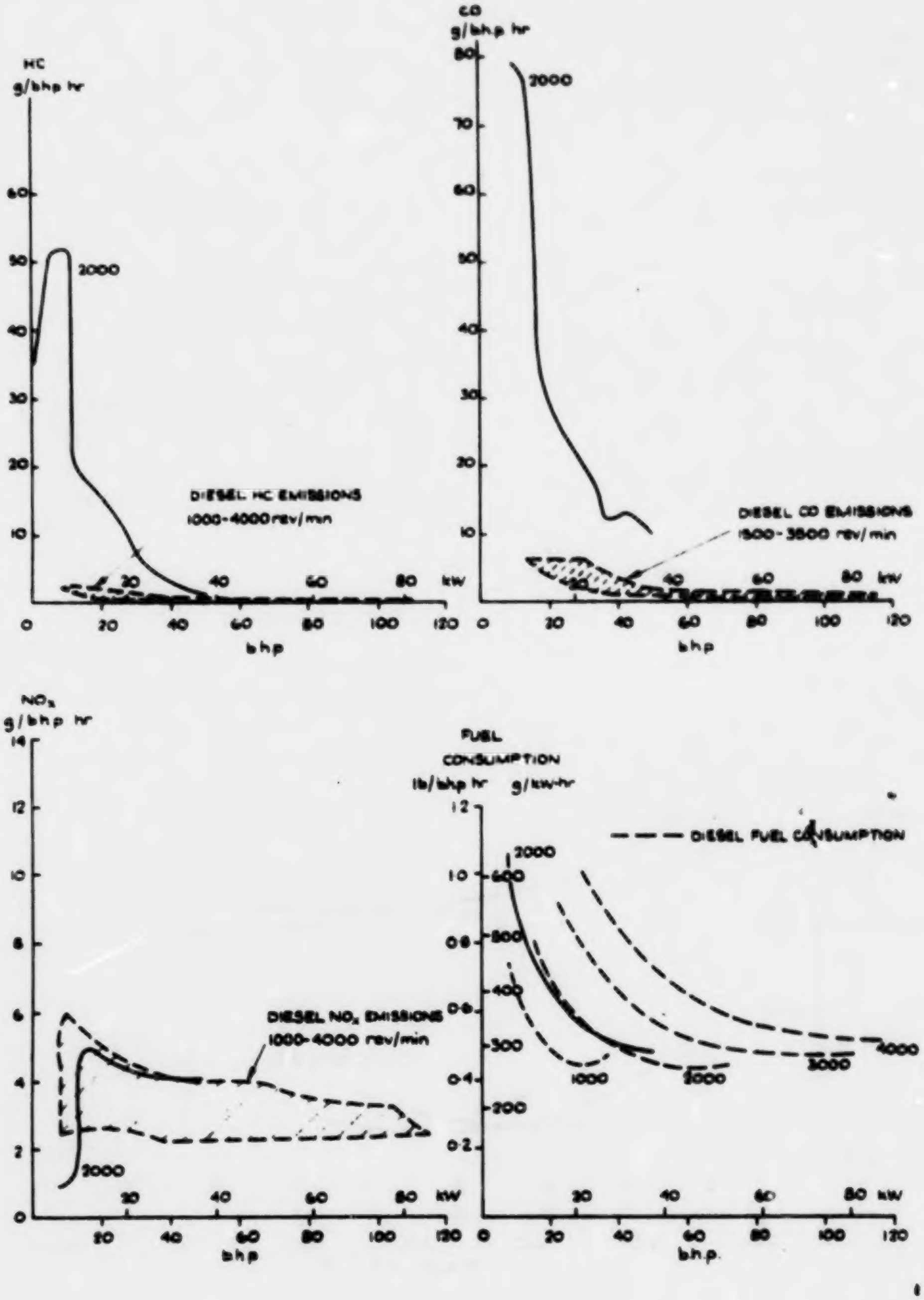


Figure 2.21. Part load fuel consumption comparisons between baseline diesel, TCCS, and Deutz.

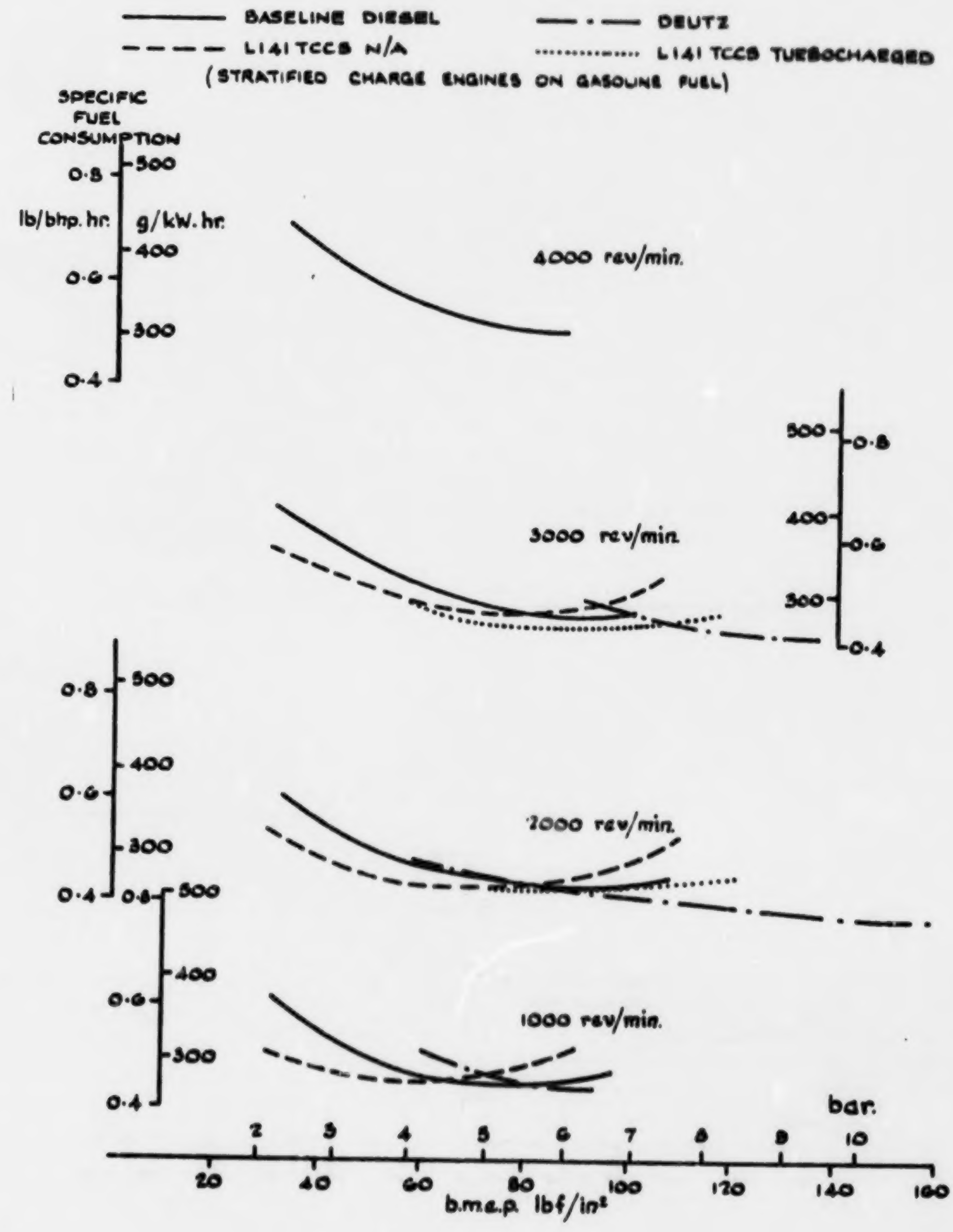


Figure 2.22. Part load fuel consumption comparisons between baseline diesel and LIS - 183 on gasoline and diesel fuels. (lis - 183 at early stage of development)

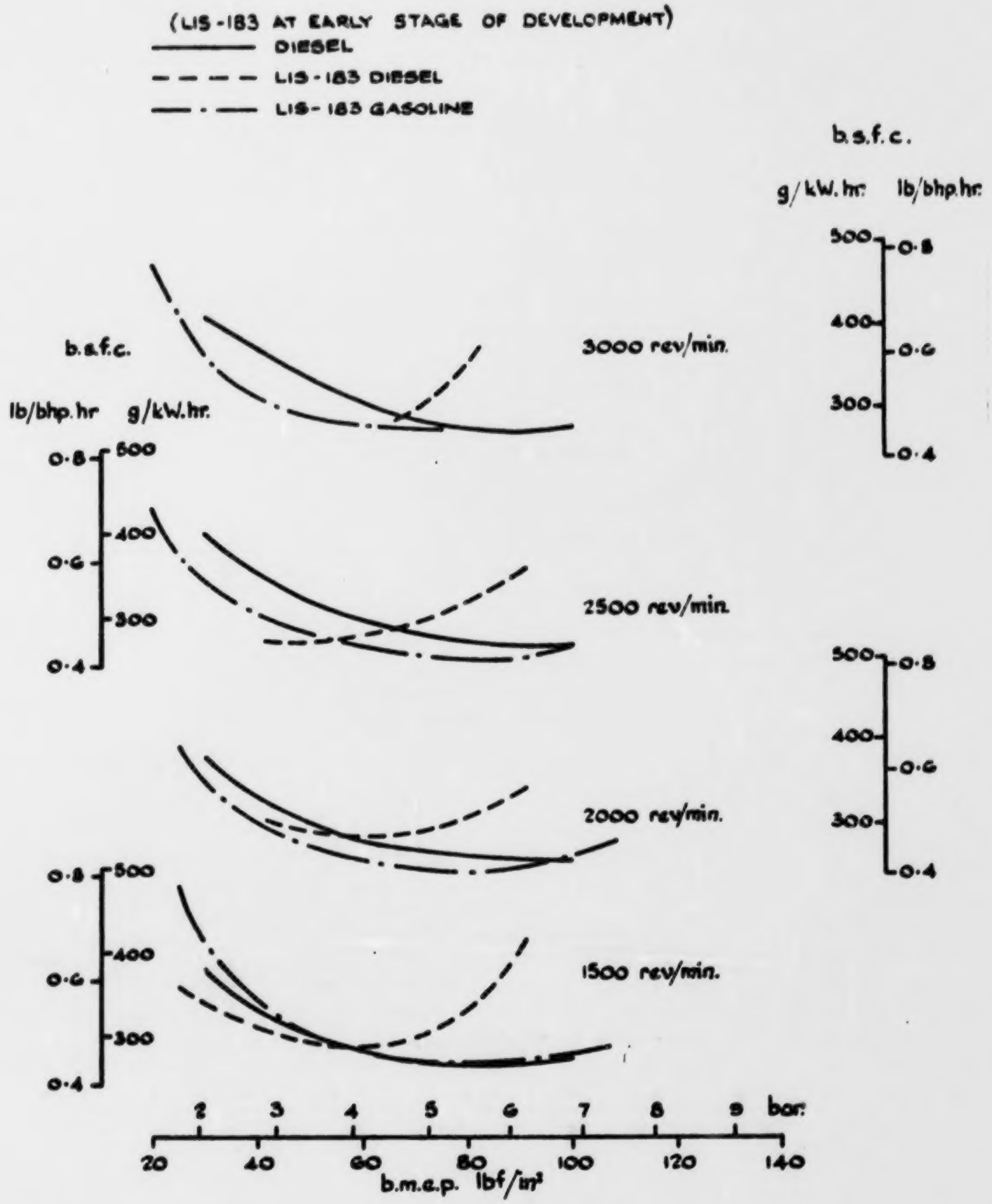


Figure 2.23. Emissions and fuel consumption of a Deutz stratified charge engine at a single point extrapolated to target horsepower.

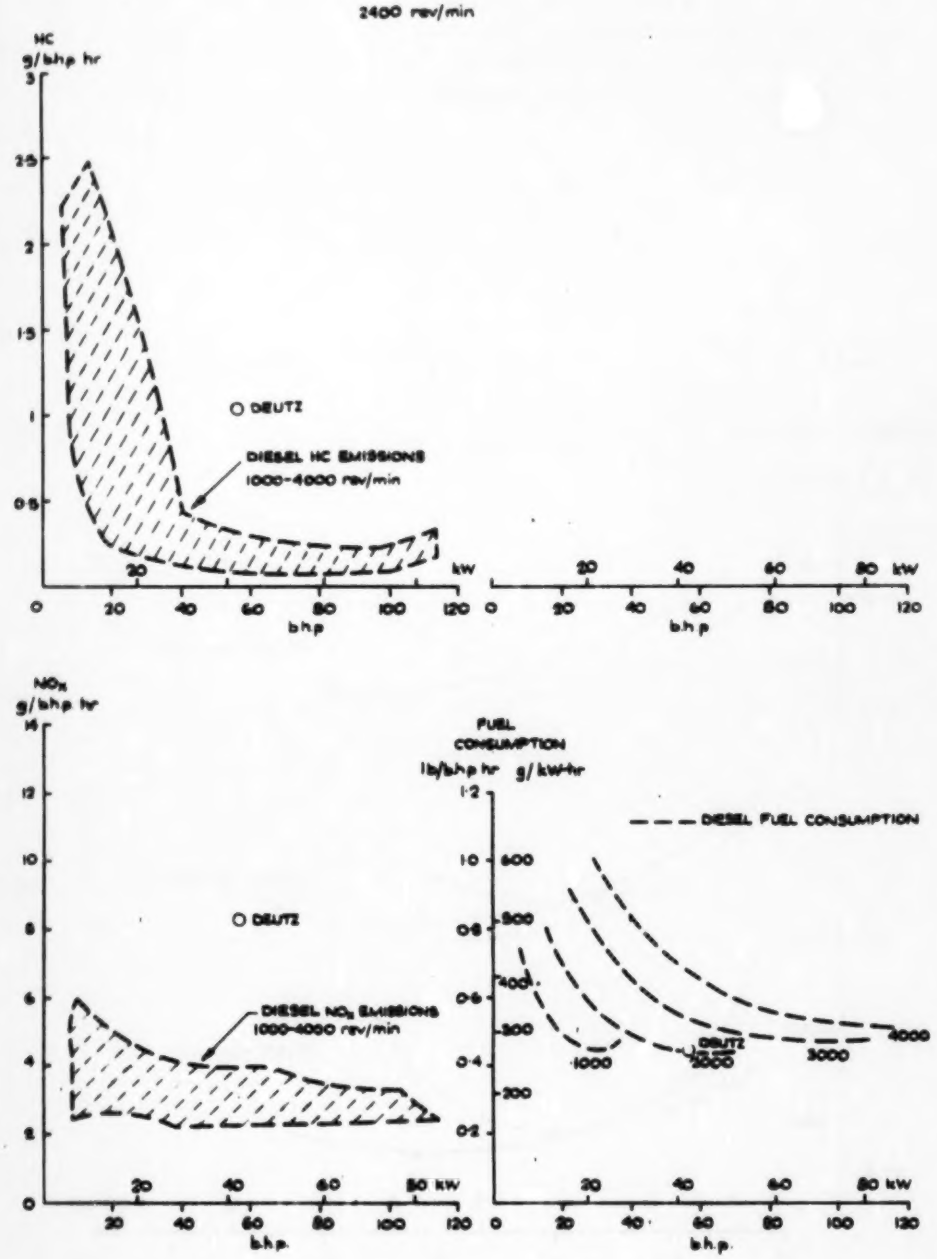
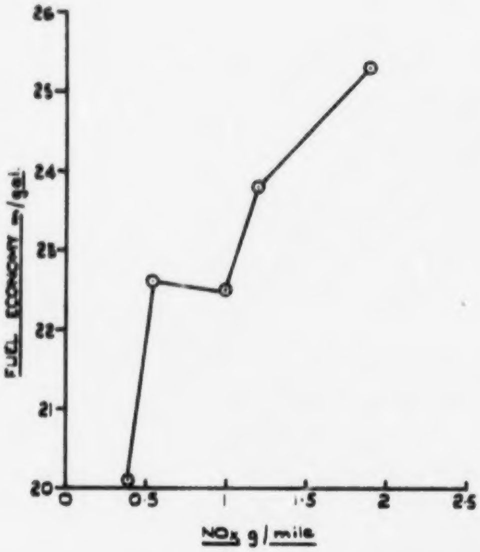
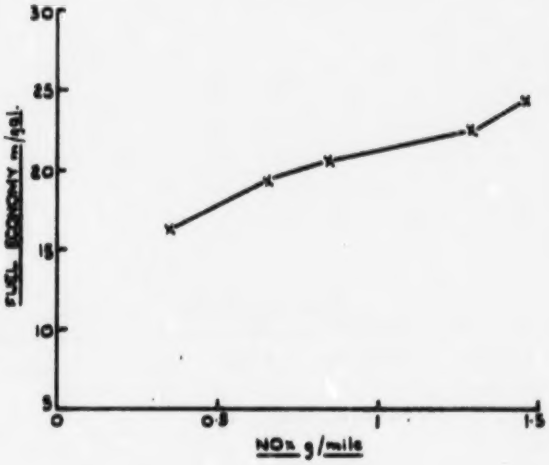


Figure 2.24. TCCS sumcompact car vehicle inertia weight 2500 pounds.



TO REDUCE NOx METHODS USED - COMBUSTION RETARD E.G.R. E.B.P

TCCS. TURBOCHARGED MISI. VEHICLE INERTIA WEIGHT 2750 lb.



TO REDUCE NOx METHODS USED-COMBUSTION RETARD E.G.R.

Figure 2.25. Effect of fuel type on economy of TCCS Cricket: plotted on basis of fuel weight turbocharged Cricket.

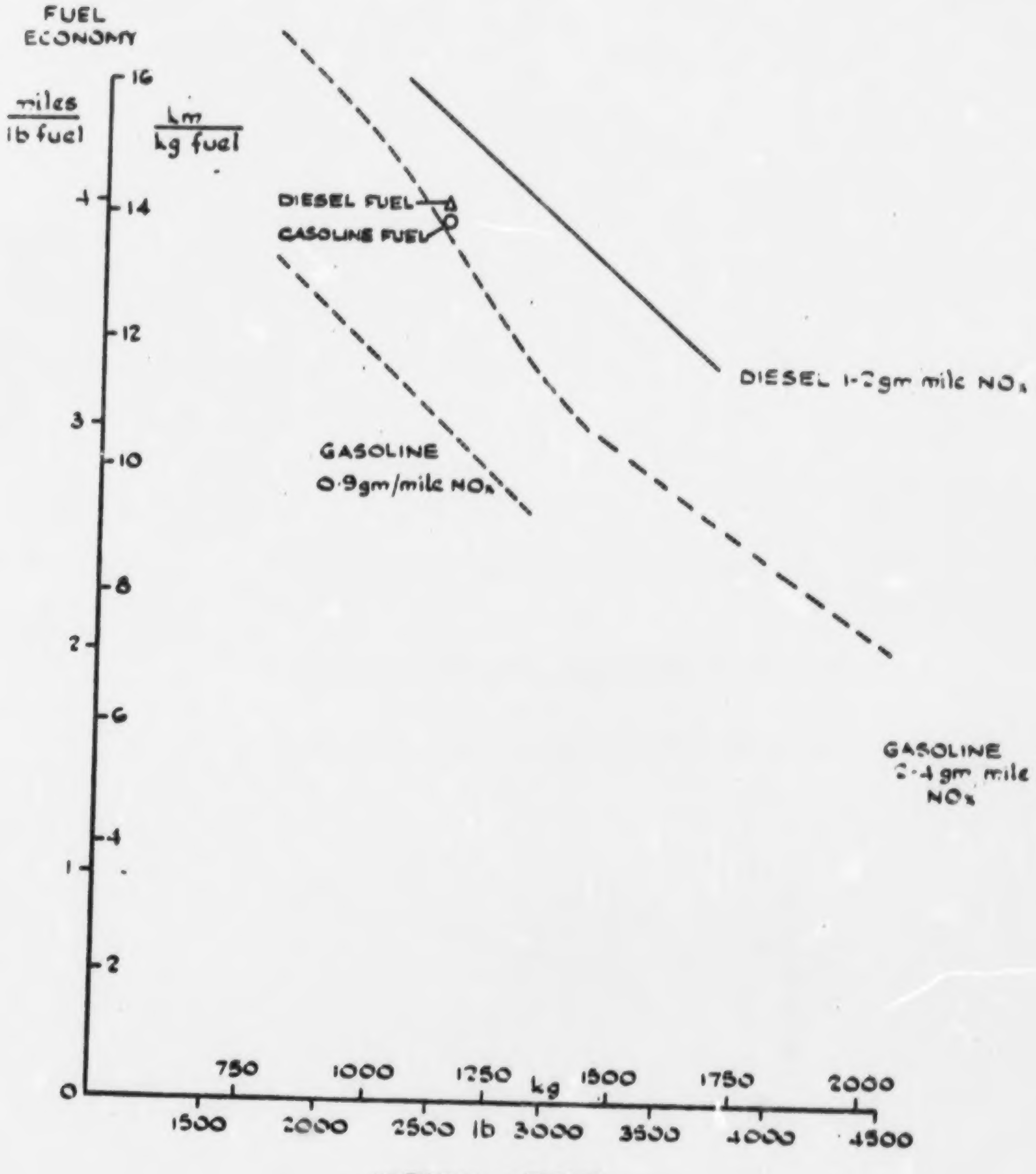


Figure 2.26. Effect of fuel type on economy of TCCS-M151 vehicle: plotted on basis of fuel weight naturally aspirated M151.

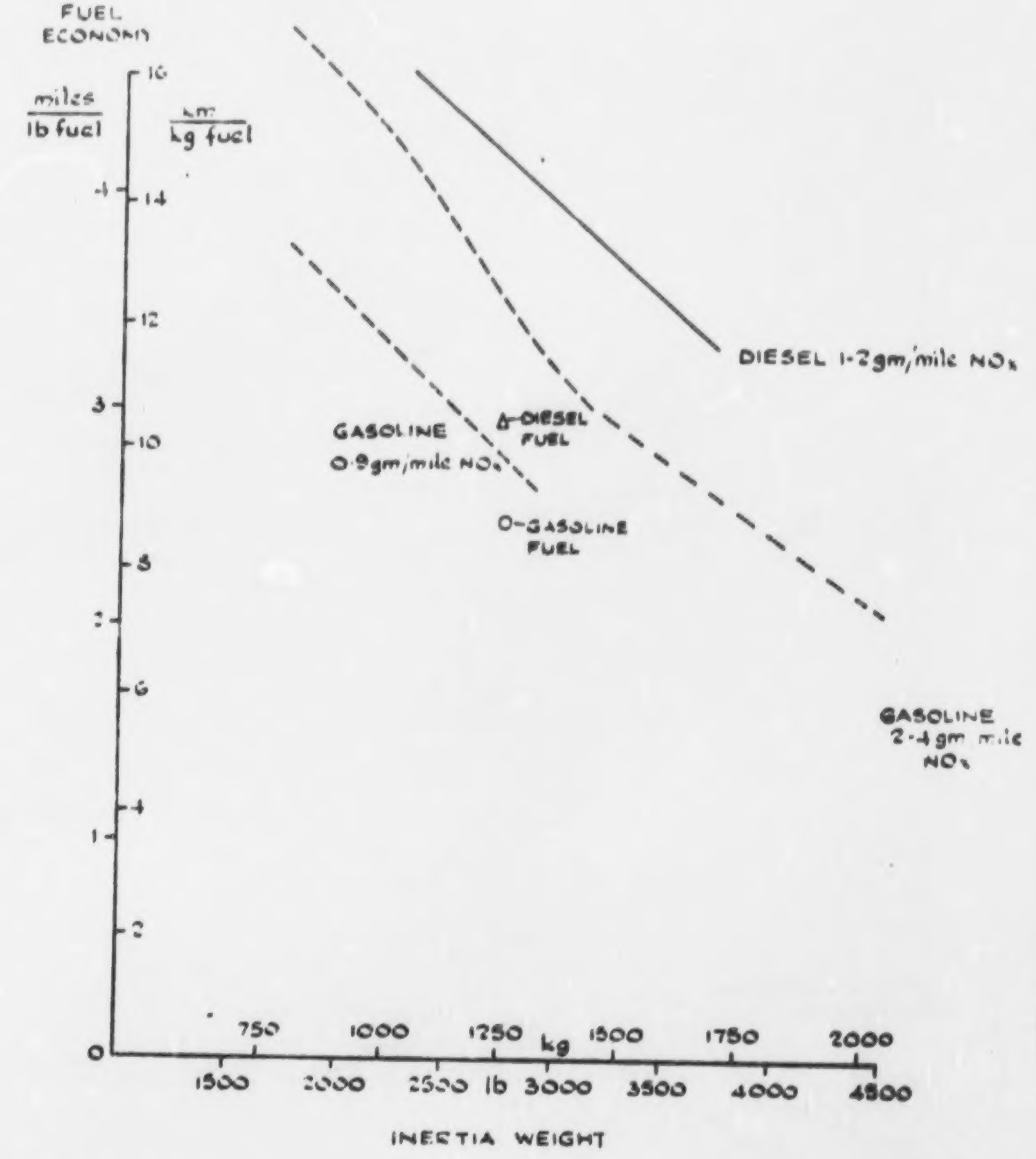


Figure 2.27. Comparison of emissions and fuel consumption of diesel engine and L141 TCCS engine no. 2350209 extrapolated to target horsepower.

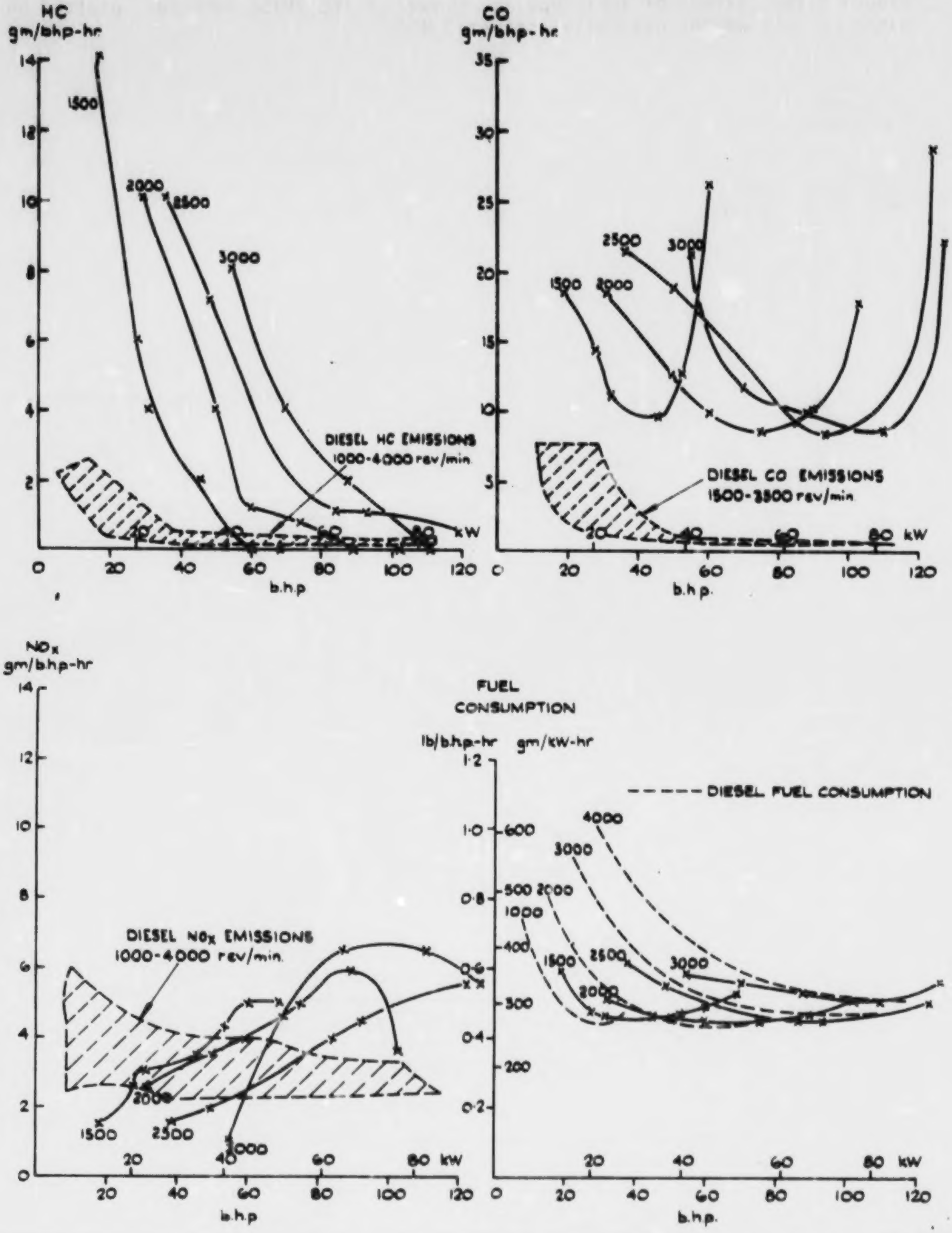


Figure 2.28. Comparison of smoke levels of various engines running on a CVS cycle. 1

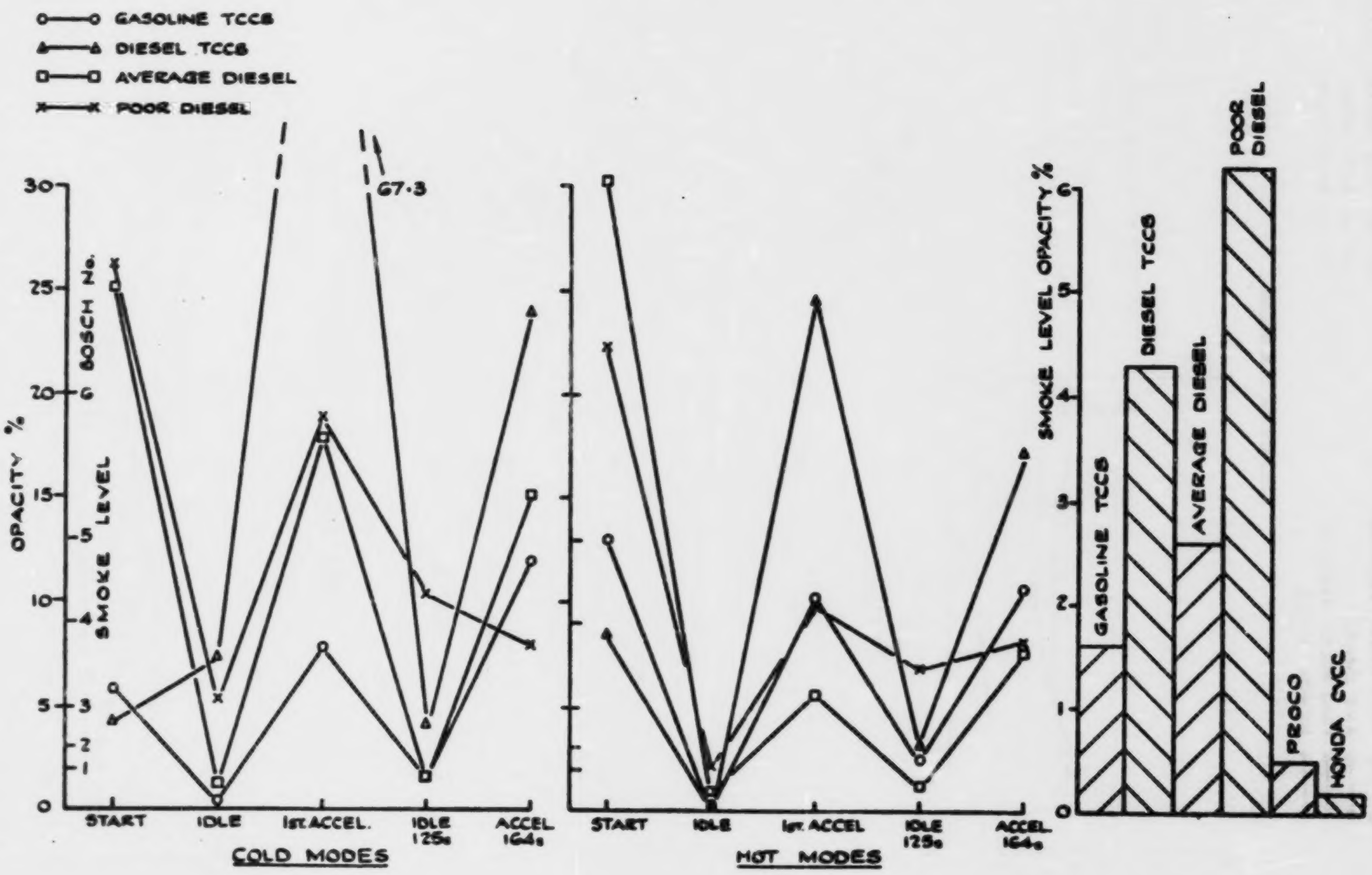


Figure 2.29. Comparison of composite odour intensity of various diesel and gasoline engines.

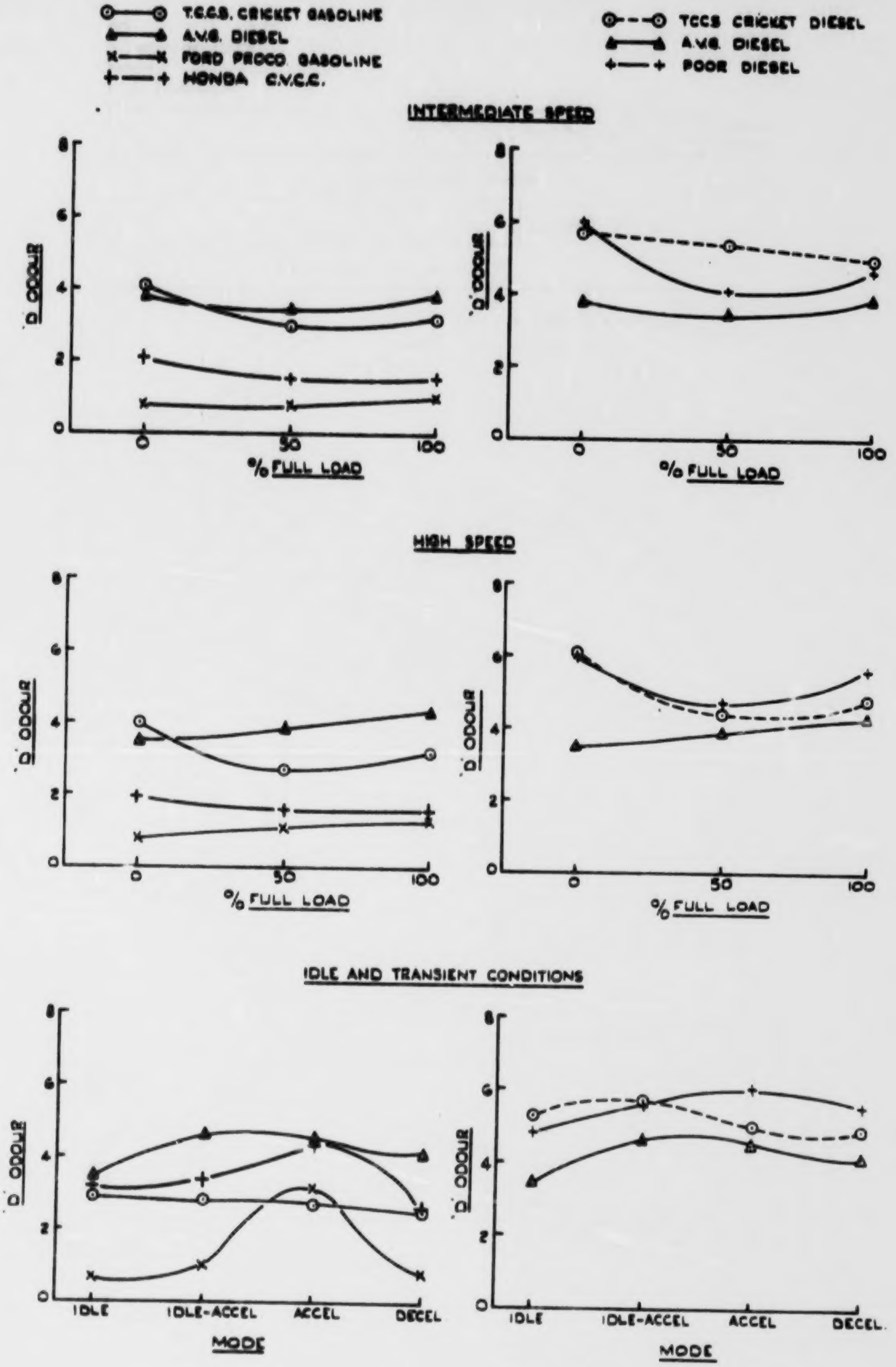


Figure 2.30. Comparison of engine weights of rotary and reciprocating gasoline engines.

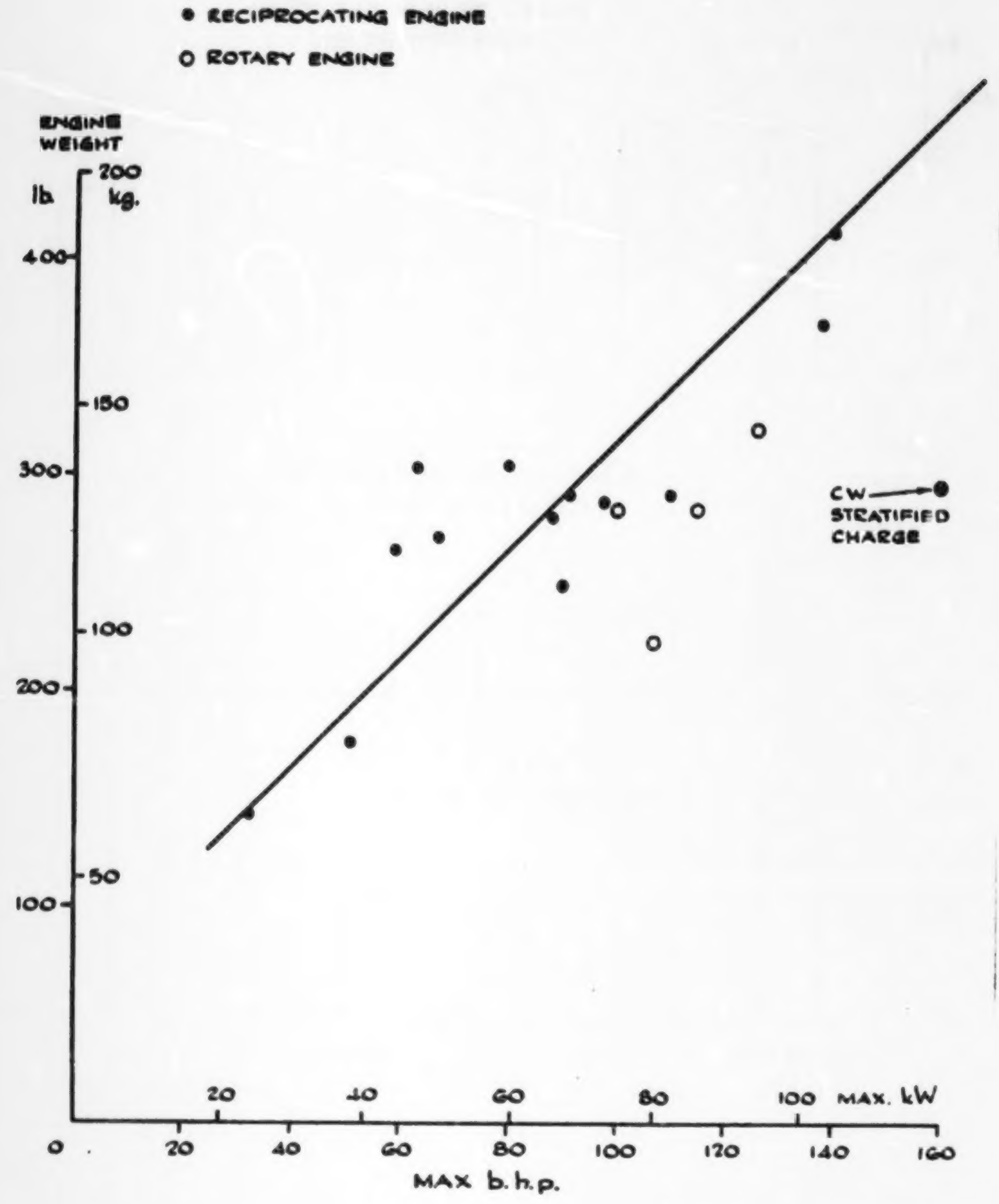


Figure 2.31. Sound level measurements on various cars, exterior measurements.

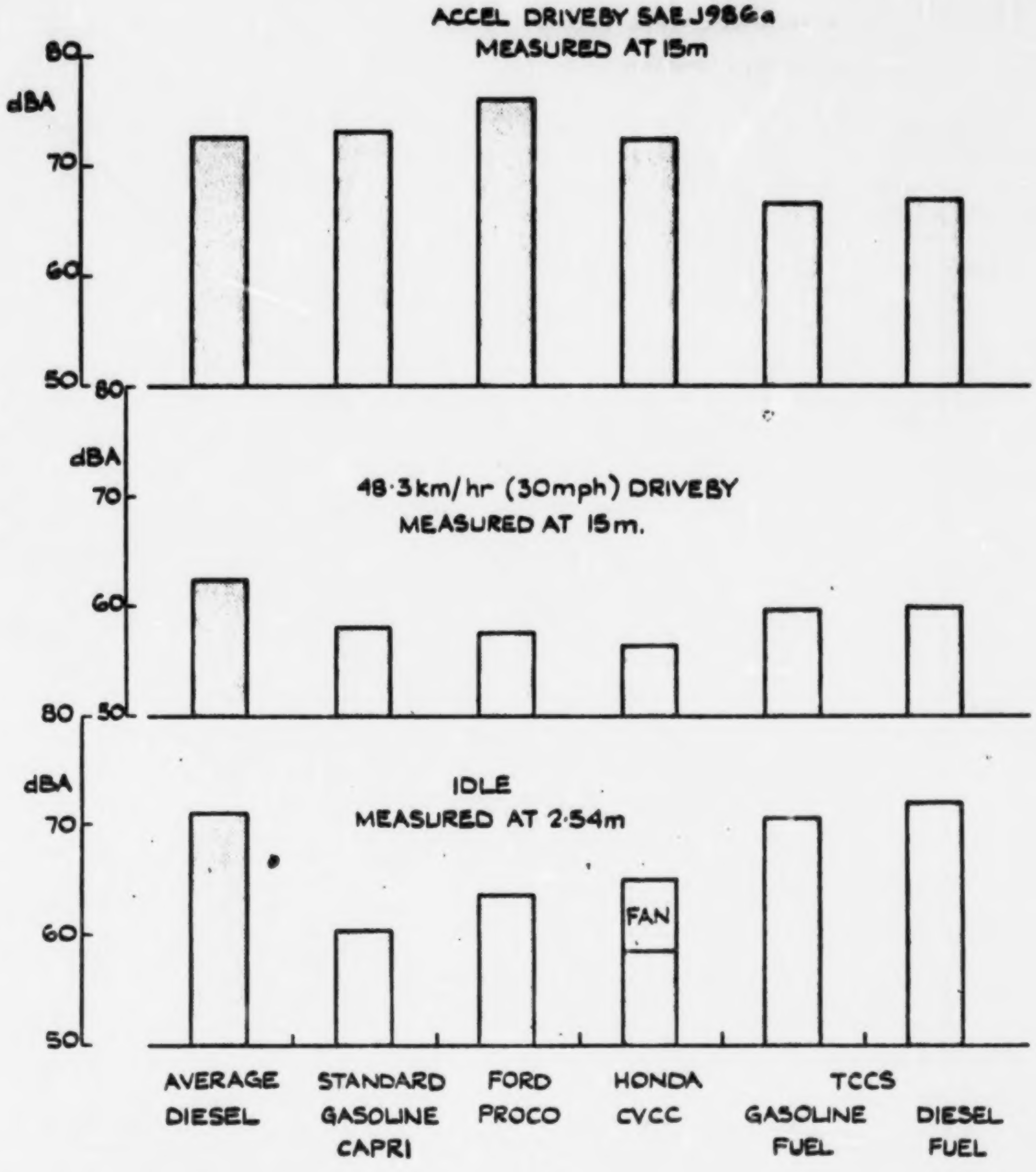


Figure 2.32. Sound level measurements on various cars, interior measurements, blower off.

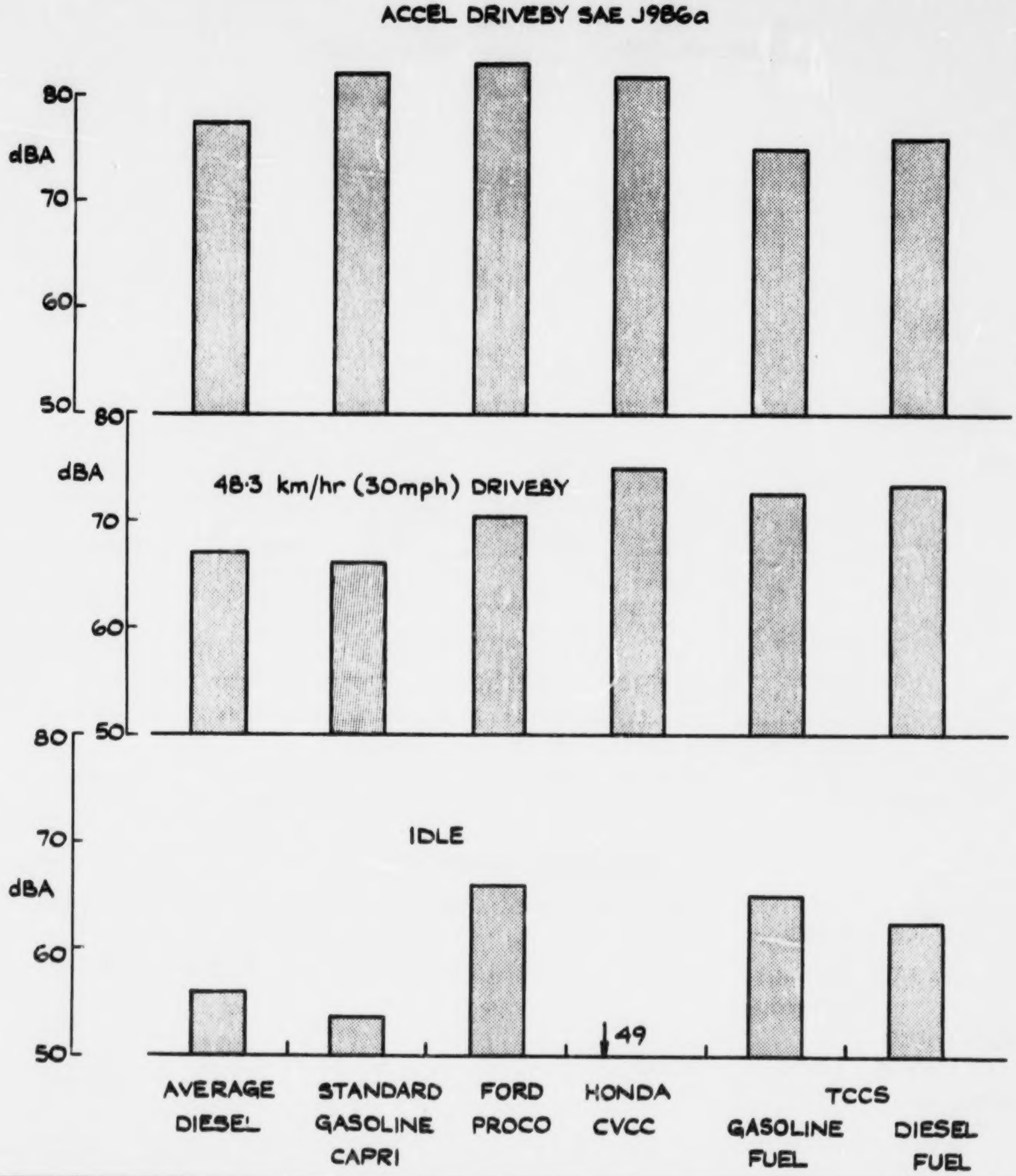
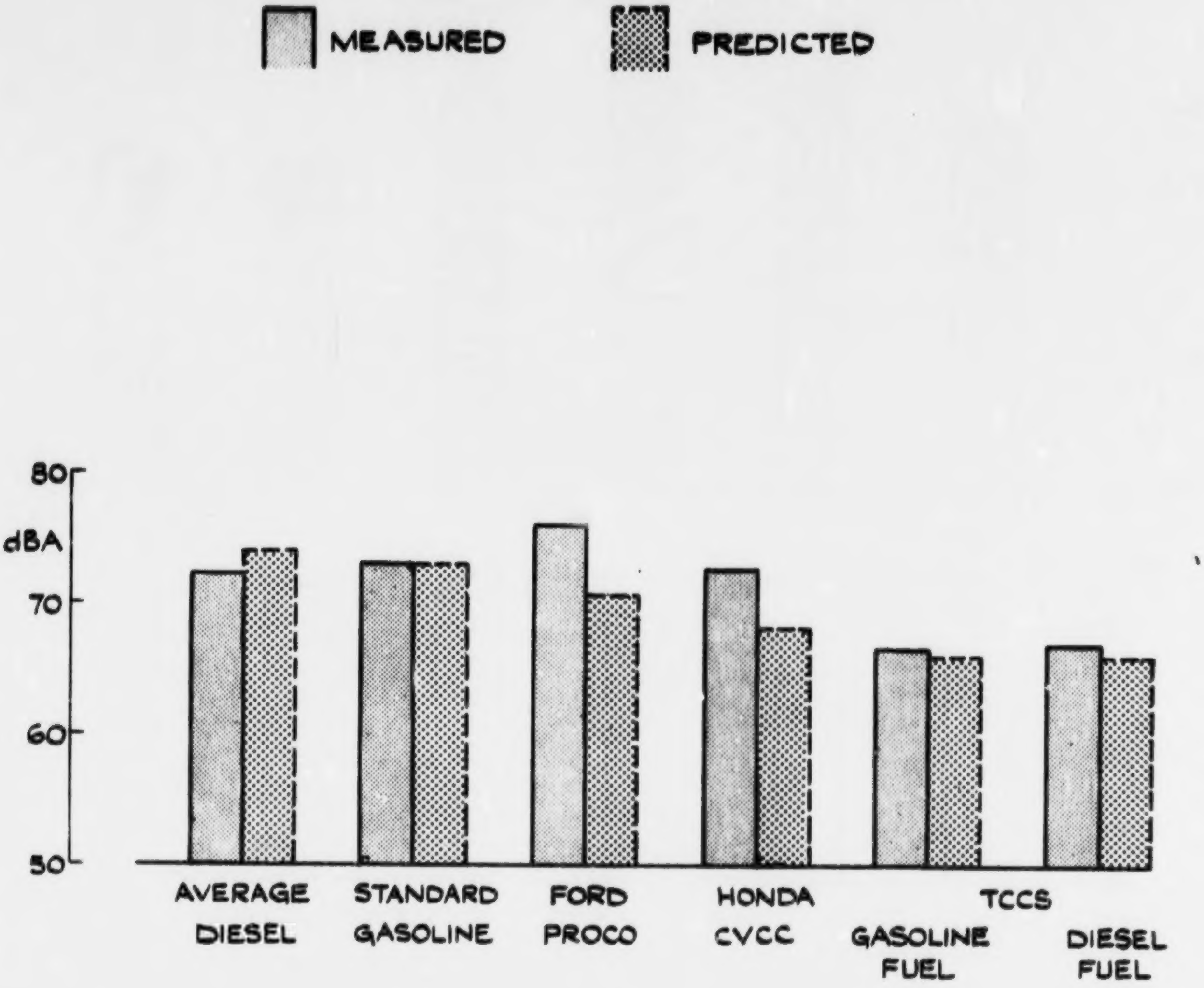


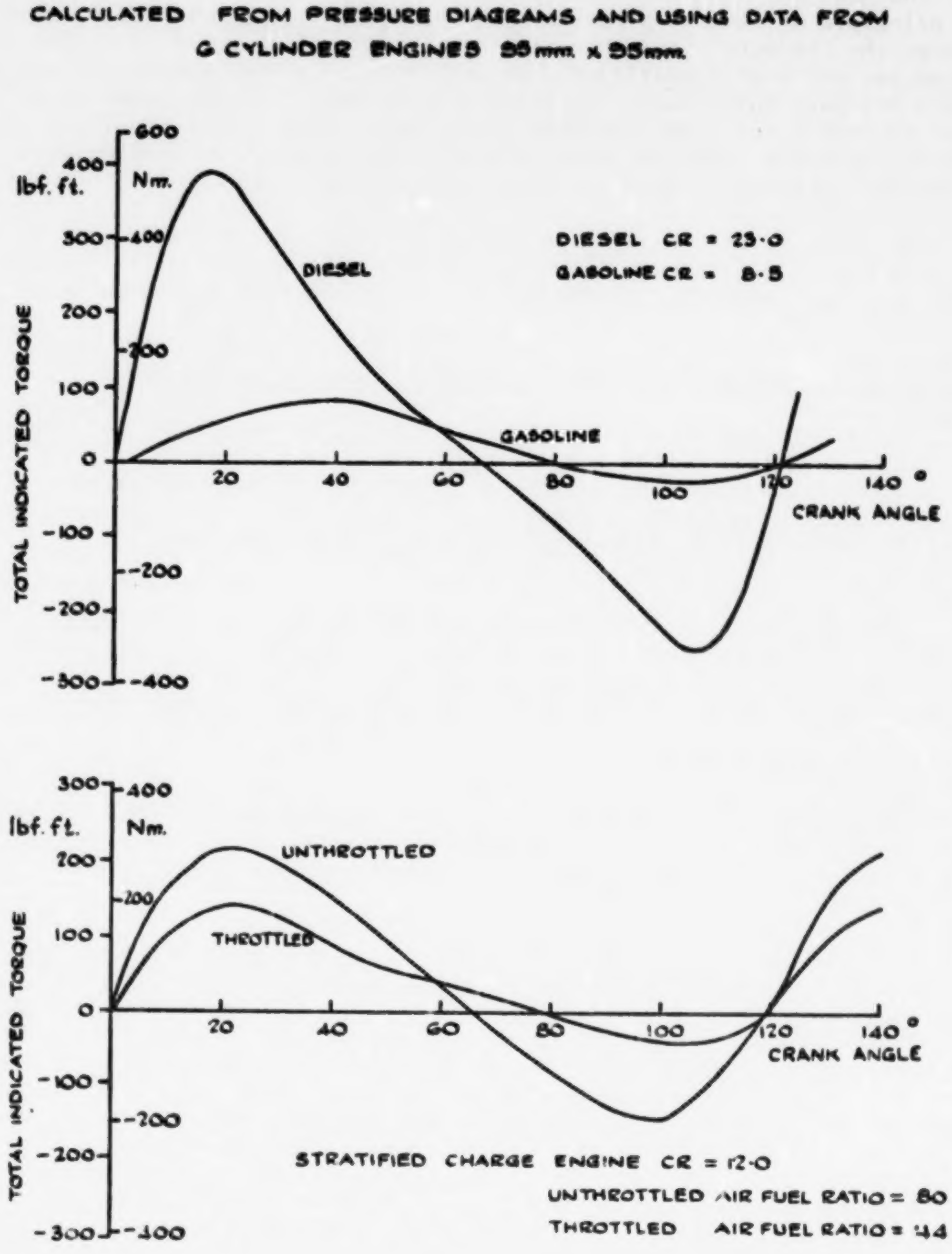
Figure 2.33. Comparison between sound level measurements and Ricardo predicted levels in SAE J986a test.



ESTIMATE BASED ON

DESIGN PRACTICE	IDI DIESEL	GASOLINE	GASOLINE	GASOLINE	GASOLINE	GASOLINE
BORE SIZE	83 to 90mm	91mm	98·4mm	74·0mm	98·2mm	98·2mm
ESTIMATED MAX. ENGINE SPEED IN TEST	4000 rev/min	5000 rev/min	4000 rev/min	5000 rev/min	3200 rev/min	3200 rev/min

Figure 2.34. Indicated total torque diagrams at 600 rev/min idle.



CALCULATED FROM PRESSURE DIAGRAMS AND USING DATA FROM
6 CYLINDER ENGINES 88mm x 95mm

CATEGORY III

SINGLE COMBUSTION CHAMBER WITH COMBUSTION CONTROLLED BY WALL EVAPORATION

History

The MAN 'FM' (FREMDZUNDUNG) system is the only one in this category because the principle of fuel evaporation from the combustion chamber wall is unique, however the characteristics are similar to those of category II. The 'FM' system was originally developed from the 'M' type diesel combustion system to enable military multi-fuel requirements to be met. It was found to be impractical to obtain self ignition when operating on high octane gasolines as the compression ratios required were very high and resulted in unacceptable mechanical stresses. Spark ignition was thus adopted.

The majority of the work carried out has been on the truck size of engines but more recently attention has been turned to the smaller sizes required for light duty applications. Table 3.1 lists the engines which have been converted.

Topic Area (a) Combustion Chamber and Stratification Technique

In the 'FM' system high levels of air swirl are generated by a helical inlet port and a spherical combustion cavity in the piston (Figs. 3.1, 3.2). During the compression stroke fuel is sprayed from a single hole nozzle onto the wall of the combustion chamber. Ideally all of the fuel should reach the wall but in practice droplets from the fringe of the spray are swept away by the air motion before they reach the wall.

The high levels of swirl help to spread the fuel over the chamber wall in a thin film and maintain good stratification of the charge. Heat supplied to the fuel from the hot piston wall and/or the charge air causes it to evaporate and mix with the air which carries it to the spark plug or flame front.

Injectors and Spark Plugs.

The injectors and spark plugs have been positioned 180° apart (included angle between plug and injector about the cylinder bore axis) in all but the most recent designs of FM engines which are being tested with the plug adjacent to the injector.

With the 180° disposition the large distance between the plug and injector results in the risk of misfire at low fuelling levels. Also long plug electrodes are required as the spray has to be angled downwards into the chamber to prevent fuel spraying onto the top of the piston and consequently by the time the fuel has traversed the 180° angle it has passed well down into the bowl. With this arrangement the spark is timed to occur after the injection of fuel is complete, and the plug must be located such that the gap is perpendicular to the air motion so that the mixture is swept through the gap.

Data is not available on the 'close' injector plug position but it will inevitably result in some differences in the combustion process. Practical limitations on the nozzle manufacture prevent the fuel being sprayed onto the chamber wall upstream (relative to the swirl) of the plug, as it is with the 180° position, hence the plug will not be fed with mixture which has been prepared by evaporation from the wall but with the mixture formed at the fringe of the fuel spray. To achieve satisfactory combustion fuel injection will probably have to be retimed to occur later in the compression stroke so that fuel is still being injected when the spark occurs. Otherwise at low fuelling levels the mixture will have been swept away by the air motion to parts of the chamber which are remote from the plug. With this arrangement, short electrode plugs should be able to be used and gap location may not be as critical as it is with the 180° spacing. However the close plug/injector spacing will result in the layout of the FM system being very similar to those in Category II and it is possible that with development the nature of the mixture formation and combustion will also become similar.

Fuel Injection.

To achieve maximum control over the combustion all of the fuel injected should be deposited on the chamber wall, so that the air motion can control the rate at which the ignitable mixture is fed to the flame front. In practice it will not be possible to prevent fuel from the edge of the spray being swept away, but to limit this to the lowest levels, atomisation and fuel spray path and envelope must be kept to a minimum.

To date conventional Bosch in line diesel fuel pumps have been used. These are capable of injecting at much higher pressures than those required when the fuel injection is completed 30° BTDC but suitable lower pressure equipment does not currently exist. If however adjacent plug/injector positions are eventually used then the later fuel injection timings required will need higher pressures to inject into the cylinder against combustion pressure and the diesel type equipment may become essential.

The injector and nozzle have to be arranged so that the fuel impinges on the wall of the chamber at a point upstream of the spark plug (when 180° position is used). To achieve this together with a satisfactory injector installation conventional diesel nozzles with an angled single hole have been employed. The fixed orifice hole type nozzles usually present problems in obtaining a satisfactory match of fuel spray characteristics to air motion over a wide range of fuelling and a pintle type nozzle with a variable orifice would be preferred, however, for this the nozzle must be co-axial with the injector and this is not possible with present injector installation required.

Current evidence indicates that a simple speed advance on the injection pump is all that is required to maintain the timing at optimum over a speed and load range.

Ignition.

A long duration spark is required to allow sufficient time for an ignitable mixture to pass through the plug gap. MAN have employed conventional Bosch coil equipment with very small plug gaps and achieved adequate duration, however durability is a problem as plug erosion shortens the period and increases the likelihood of misfire. No advance of ignition timing with speed or load has been provided on any of the FM engines so far and satisfactory operation has been obtained, with the timing fixed between 10 and 15° BTDC (to give a start of combustion at 5° BTDC).

With conventional ignition equipment the spark will occur for a fixed time and hence the duration in engine crank degrees will increase with speed. An ignition system which enables the spark duration to be controlled to a fixed crank period over the speed range should result in increased plug life.

TABLE 3.1 FM-TEST ENGINES

TYPE	No. OF CYL.	BORE × STROKE mm mm	RATED SPEED rev/min	COOLING	REMARKS
A	1	105 × 109	3200	Water	
B	1	92 × 100	3800	Air	
C	1	120 × 125	2800	Air	
D	1	120 × 124	3200	Water	Turbocharged (simulated)
E	1	90 × 95	5000	Water	Tests started with max.4000 rev/min.
F	8 V-type	114 × 109	3200	Water	
G	8 "	120 × 125	2800	Air	
H	6 "	120 × 125	2800	Air	Turbocharged
J	12 "	125 × 130	2500	Water	Turbocharged
K	4 in-line	92 × 100	3800	Air	
L	6 "	108 × 128	2500	Water	
M	6 "	112 × 140	2200	Water	
N	12 V-type	120 × 125	2800	Air	Turbocharged

Compression Ratio.

The compression ratios used have been in the range 15.5 to 17.0 to 1. These may have been inclueneced by the multi-fuel requirement as the engines were required to operate on diesel fuel without spark assisted ignition.

As compression ratio is increased indicated thermal efficiency improves but mechanical friction and stresses rise. The balance between these conflicting factors occurs around 14 to 1. The main limitation to an FM system being built with this ratio would be the maximum combustion bowl depth which can be fitted to a given engine design.

Swirl.

Very high swirl levels are required in the FM system. They are generated by a helical type inlet port and the spherical combustion bowl.

Measurements indicate that at the end of the induction stroke the swirl levels are 3.5 x engine speed and that by the end of compression this has been magnified to 10-15 x engine speed.

The large increases of swirl during compression are achieved by the relatively small bowl diameter (c. 45% of bore) however as bore size decreases so installation problems are encountered with the injector and spark plug because they move very close to the cylinder centre line and interfere with the ports. Also the cut-outs in the piston cavity for the plug and injection become a large proportion of the chamber volume and will begin to reduce the swirl levels.

Throttling.

Throttling has not been employed on any of the FM multi-fuel engines but the possibility of introducing it as a means of controlling hydrocarbon exhaust emissions has been considered.

Topic Area (b) Combustion Characteristics and Heat Rejection

The cylinder pressure diagrams which are available for the FM system show that smooth stable diagrams are achieved with most fuels. The exceptions are low octane gasolines (RON c. 80) on which knocking can occur. Low octane heavier fuels such as JP4 and diesel fuel however do not knock. This is presumably due to differences in the mode of mixture formation and combustion. The amount of fuel prepared before ignition on the high volatility gasolines will be greater than with lower volatility fuels such as diesel and JP4 and hence the latter fuels, in spite of their lower resistance to knock can give smooth diagrams. Examples of the pressure diagram obtained when operating on 93 RON gasoline are shows in Fig. 3.3.

References 3.13 and 3.1.9 give comparisons of measured heat release diagrams with those calculated assuming that combustion is controlled by the rate of evaporation of the fuel film. These show that in the case of diesel fuel good agreement is achieved but with gasoline the agreement is not as good, presumably because the higher volatility results in more fuel being evaporated from the edge of the fuel spray prior to it reaching the wall.

Calculations indicate that the combustion chamber wall temperature appears to be important in the initial and final stages of mixture formation as it will be above that of the fuel film and hence have a significant affect on the rates of evaporation. During the main combustion phase however, the wall temperature would appear to have little effect as the data indicates that the film temperature is above that of the wall.

Typically the timing of events for an FM engine will be as follows:

Injection	Complete before ignition
Ignition	10-15° BTDC
Start of Combustion	5° BTDC
Pmax	10°-20° ATDC

The delay between ignition and start of combustion is quite short indicating high levels of turbulence within the chamber.

Heat rejection to the coolant of the FM is believed to be the same as for the 'M' system which for Truck type engines is 70-75% of the output power. Allowing for size effect this would be expected to rise to around 100% for smaller light duty engines.

Topic Area (c) Performance

The principal factors which will affect the power output of FM engines are:

1. Volumetric efficiency - The high swirl port which is required should be capable of returning good volumetric efficiency over the mid speed range but at high speeds the restrictive nature of the ports will cause the air flows to drop rapidly. To minimise this effect long valve opening periods can be employed but this will compromise the efficiency which can be obtained at low speed. MAN have shown that turbocharging of the FM system is possible and, if required can be used to compensate for the poor high speed air flows.

MAN also have some evidence that lower swirl levels can be used for gasoline operation and hence an improvement in high speed volumetric efficiency achieved if there is no multi-fuel requirement for the engine.

2. Maximum Speed - The maximum rotational speed that an FM engine has been operated at is 4000 rev/min, however data is only available for the 95 x 100 mm military engine operating at 3800 rev/min. The rapidly decreasing volumetric efficiency will limit the maximum useful speed to which the engine can be run and present data suggest that this is around 4000 rev/min.

3. Smoke Level - The FM power output is limited by acceptable smoke, with gasoline the data indicates that for smoke levels in the range 1 - 1.5 Bosch the air fuel ratio will be around 19 to 1, falling to around 17 to 1 for a smoke level of about Bosch 3. On diesel fuel the air fuel ratio at Bosch 3 will be close to 18.5 to 1. This results in about 10% greater output from gasoline than diesel fuel for a given smoke limit, due to the lower smoke formation properties of gasoline.

Test Bed Results.

A torque curve extrapolated to the target horsepower from the FM L9204 engine is shown in Fig. 3.4 and compared to the base line diesel engine. This shows that almost equal swept volumes are required for the FM and diesel and that at high speed the torque levels are similar. However the FM operates over a narrower speed range and the torque figures below 2500 rev/min are much worse than the diesel. It is possible that the shape of the curve of the FM engine can be attributed to the military application for which it was developed, so the torques which might be achieved at any speed for a fixed smoke limited A/F ratio were calculated. The results indicated (Fig.3.4) that the curve shape might be similar to or better than, that of the diesel depending on the fuel used and the swept volume required to achieve the target horse power. The main factor which might prevent these torque curves being achieved will be the problem of matching fuel injection to air motion over the wide speed and load range. Undoubtedly compromises which will affect the curve shape will be necessary but the magnitude of the effect is a matter for experimental investigation.

Turbocharging data are only available for engines of considerably larger bore size and more limited speed range than those required for automotive applications. It would be unrealistic to extrapolate or interpolate these data to the target horse power but Fig.3.5 shows the bmep curves obtained from the turbocharged engines. One achieves a peak bmep of 12.2 bar and the other 9.4 bar due to different boosting levels. For comparison purposes the bmep curves for two naturally aspirated engines are also shown.

Vehicle Tests.

The L9204 FMV engine has been fitted to a light commercial vehicle but no performance data are available.

Topic Area (d) Fuel Consumption

The fuel consumptions from a range of FM engines operating on gasoline are plotted in Figs.3.6 to 3.8 (Ref.3.1.8). From these it will be seen that the general levels are very good although the data are limited to only one engine at 3500 and 3750 rev/min. It is worth noting that the engine which operates at the highest speed has fuel consumption some 5 - 10% worse than the other engines at 1750 rev/min, indicating that there may be problems in maintaining good consumption over a wide speed range. Data from the L9204 (engine K) air cooled engine has been extrapolated to the target horse power and the results plotted in Fig.3.9. Here the good fuel consumption levels are confirmed as they are better than those of the base line diesel at all of the conditions compared.

The high levels of brake thermal efficiency which are achieved can be attributed to a combination of good cycle efficiency resulting from compression ratios of around 14 to 1, which must be close to the optimum for these engines, and friction levels lower than the diesel engine. Fig.3.10 which shows the friction data (Reference 3.1.9) from one engine indicates that the levels are about halfway between the conventional diesel and gasoline units. With the scope which exists for controlling the peak cylinder pressures by adjustment of injection and ignition timing, friction might be reduced further during development by lowering mechanical loadings.

The fuel consumption loops for both diesel and gasoline have very similar shapes with almost identical minimum points but the loops 'turn up' at a lower load on diesel fuel.

The results available for the two turbocharged engines show that in the case where the output has been increased by c. 25% (engine J fig.3.7) the fuel consumption over the light load part of the loops are similar to those under naturally aspirated conditions and that the minimum consumption is also similar in value but continues further up the load range. The engine which is turbocharged to c. 50% (engine D fig.3.6) however has similar minimum consumption to the naturally aspirated engine but considerably worse light load values (Figs.3.6 and 3.7). This would almost certainly have a significant adverse affect on vehicle fuel consumption, should 50% turbocharging be required and the same characteristic exists on light duty engines.

Vehicle Results.

No actual vehicle fuel consumption results are given for the FM but mention is made in one reference (3.1.7) that for a light duty truck, the L9204, gave a 30% improvement in fuel consumption over that of the same truck fitted with a conventional gasoline engine. Certainly from the consumption loops which have been obtained, this improvement would seem reasonable providing any adjustments required to achieve acceptable driveability do not have an adverse affect on fuel consumption.

Topic Area (e) Emissions

Only a limited amount of emissions data were available for the FM system. These related to one single cylinder engine with the results quoted in grams of pollutant per kg of fuel. These figures were adjusted to multi-cylinder values taking typical specific fuel consumptions and then adjusting them to the target horse power. The results are plotted in Figure 3.9.

It will be seen that compared to the base line diesel the NO_x and CO values are similar over the load and speed range but that the HC levels are high, particularly at light loads. However these HC levels are exceptionally low when compared with other stratified charge engines and follow a general trend observed in the survey that as compression ratio is increased, so HC levels fall. The unburnt hydrocarbons are caused primarily by the fuel which is swept away from the fringe of the fuel spray to form areas in the chamber which are too weak to burn and although the flame quenching effect of the chamber wall will also have an effect, this is considered by MAN to be a secondary factor.

The reduction of HC levels by inlet air throttling has been investigated and the data are shown in Figure 3.11 together with the penalty to fuel consumption which is incurred. It will be seen that a 25- 0% reduction can be achieved over the speed range with about 50% throttling but fuel consumption rises by only 5-8%. This degree of throttling also reduces NO_x levels by between 30-50% but inevitably some increase in CO levels occur, although these are only around 10%. MAN have some evidence that moving the plug closer to the injector may help to reduce HC levels but the data is only of a preliminary nature.

There are no emissions data available for vehicles fitted with an FM engine and hence the prediction of whether the primary or secondary targets are likely to be met is difficult. The base line diesel shown in Figure 3.9 will meet the primary targets for HC, CO and NO and hence the data for the FM are well within the diesel band for NO_x and close to it for CO, it is felt that an FM engine would meet the required levels. HC levels are, however, well outside the diesel band and are likely to increase above those shown with reductions in cylinder size. Also during cold starts the HC emissions are likely to be high due to the low temperature of the combustion chamber wall. In an uncontrolled form an FM engine would not meet the 0.41 g/mile limit but with the use of throttling and/or a catalyst the target should be able to be achieved.

The secondary targets for NO_x stand a reasonable chance of being met with exhaust gas recirculation although the reduction in power output which would result cannot be predicted. Also, as E.G.R. will result in increased HC, further work may be required to keep the emission within the limit.

Smoke.

As discussed in the Performance Section (c), the power output will be limited by acceptable smoke levels but problems can also occur with smoke emissions at part load. On the 'M' system diesel chamber blue/white smoke is often emitted at light loads and this characteristic also occurs on the FM system when operating on diesel fuel. The problem is associated with poor evaporation of the fuel from the piston wall due to low metal temperatures and hence poor mixture formation. Insulation of the chamber cavity may alleviate the problem and MAN are investigating this as a possible solution for the 'M' system.

Odour.

The limited data available on the odour from FM engines indicates that at light loads levels are unacceptably high for the light duty automotive applications. The situation is aggravated by the odour being highly irritant to the nose and eyes. Presumably detailed development will enable smoke reduction to be effected but the amound of reduction which can be achieved cannot be predicted. MAN have preliminary data which indicates that moving the spark plug close to the injector may result in reduced levels of both odour and hydrocarbons.

Particulates.

No data are available on particulate emissions but they would be expected to be the same order of magnitude as a diesel.

Topic Area (f) Multi-Fuel Capability

The multi-fuel capability of the FM system has been clearly demonstrated (3.1.7/8/9/10). Gasoline with octane numbers from 80-100, JP4, diesel fuel, diesel-gasoline mixtures and methanol have all been used, and satisfactory operation and starting was achieved. An adjustable maximum fuel stop was required to allow for the different specific gravities of the fuel but injection timing does not have to be altered providing the timing is set for optimum on diesel fuel and a late start of combustion is accepted on gasoline operation. This should not affect the power output rates of a multi-fuel engine since this is set by the smoke limit on diesel fuel. Gravimetric fuel consumption on all fuels is about the same but the smoke limited power output will vary considerably, diesel fuel giving some 10% less than high octane gasoline. The literature indicates that with methanol the output can be increased a further 10% without problems, although the reduced calorific value means that the volume of fuel which has to be delivered is about twice that of gasoline for a given power output.

Topic Area (g) Engine Components

Ignition.

The long duration spark which is required to ensure that combustion is initiated has led to spark plug durability problems in the engines tested to date. Electrode erosion shortens the spark duration which then leads to misfire. The use of platinum tipped electrodes gave a significant improvement in life over Nimonic during a Nato test cycle. The Nimonic plugs reached the limit of acceptability after 190 hours but the platinum plugs achieved 540 hours.

Current plug positioning requires the gap to be located and hence conventional screw-in type plugs are not suitable.

Injection Equipment.

Conventional Bosch in-line diesel equipment has been used to date but scope exists for employing lower pressure equipment not only in the interests of costs but also to minimise fuel atomisation. Currently there are no fuel injection pumps being manufactured in production quantities which operate at the intermediate pressures below that of the diesel but above that of the port injected gasoline engine. Scope obviously exists for the development in this area.

Other Components.

The basic FM engine components will be similar to those of diesel and gasoline engines. The piston will be heavier and more complex to machine as they have a re-entrant cavity with cut-outs for the injector and plug. Also the deep bowl means that cylinder block height may be greater than many conventional gasoline engines.

The cylinder head will be more complex than the gasoline engine but slightly less so than the indirect injection diesel as it has no pre-chamber, however location of the plug and injector near to the cylinder bore centre line will need careful detail design if adequate cooling is to be maintained in the valve bridge region of the head.

Topic Area (i) Mathematical Models

As with Category II engines, computer modelling of the FM type of engine is extremely complex because both liquid and vapour fuels are distributed unevenly in the combustion chamber.

One reference (3.1.3) indicates that MAN have a simplified model for 'M' and 'FM' systems where the fuel is being evaporated from the wall. The results show that good agreement was obtained between measured and calculated heat release diagrams for diesel fuel but not with gasoline.

Topic Area (j) Engine Size and Weight

Reference 3.1.10 is the only one which contains data on FM engine size and weight. The information is not directly comparable with that from conventional engines as the unit referred to is a specially designed air cooled military unit and the figures quoted are for the complete engine package including fan and cowling.

length	93.8 cm
width	60.8 cm
height	68.6 cm
weight	210 kg

The engine is a 2.66l 4 cylinder unit producing 52 kW at 3800 rev/min.

Topic Area (k) Noise and Vibration

The cylinder pressure diagrams shown for FM engines are all very smooth with rates of pressure rise in the region of 2-3 bar/crank°. Peak cylinder pressures have been about 70 bar (1000 lb/in^2) but it has been shown that scope exists for controlling this with variation of the injection and ignition timing although fuel economy inevitably suffers, but to a degree which is not specified. There are no measured noise levels published but at full load conditions they would be expected to be fairly low because of the combined effect of smooth pressure diagram and limited speed range. It is reported by MAN that at idle combustion noise is also low and that the sound of the injector equipment predominates. However, the peak pressures in the unthrottled FM will be higher at part load conditions than in the gasoline engine and hence probably give higher noise levels. Instantaneous torque fluctuations between the engines and drive train will also be greater and as discussed for Category II engines, benefits could be derived by throttling.

Topic Area (l) Durability & Starting

The L9204 engine was subjected to both long-term dynamometer runs and extensive road tests to assess the endurance and reliability of the engine (3.1.10). The road tests cover a wide range of conditions including temperature variations from -22°C to 42°C and operation at altitudes up to 2300 m.

The literature does not give details of engine conditions at the completion of the endurance tests but implies that no major problems were encountered. Apart from spark plug durability this is in line with experience on diesel engines which have good durability in spite of higher mechanical and similar thermal loadings to an FM engine.

A starting aid was fitted to the L9204 engine for cold starting in the diesel mode but this was not required for gasoline operation. No details are given on the temperature range over which the engine can be started on diesel fuel without aid.

Application to Motorcycles

The same comments as made in categories 1 and 2 as to the general unsuitable nature of this application also apply to this section. The MAN-FM combustion process has been applied to air cooled engines, but in cylinder sizes greater than 600 cc.

Conclusions

1. The FM engine is the only system which comes into the category where combustion is controlled by evaporation of the fuel from the chamber wall.

The literature survey has indicated this system can give extremely good test bed fuel consumption on a wide range of fuels including gasoline, JP4, diesel fuel and methanol.

2. With a catalyst to give adequate hydrocarbon control, the system should be capable of meeting the primary emissions target. Exhaust gas recirculation should enable the secondary target to be met although experimental work would be required to establish this.

3. Power output will be set by the smoke levels and speed range; the maximum rotational speed being limited to around 4000 rev/min by the flow characteristics of the high swirl inlet port which is required. For a given swept volume output will be similar to a diesel - some 70% of that of a conventional gasoline engine.

4. The test bed fuel consumption and exhaust emission results from the FM system are the best of the direct injection stratified charge systems which have been covered by this survey.

5. The development of the FM system has not been directed towards light duty automotive engines and a number of possible problem areas have been identified for this application. (1) It may be difficult to obtain a satisfactory match of fuel injection characteristic and air motion over a wide speed range. (2) Smaller bore sizes may result in increased HC emission. (3) HC emissions under cold start conditions may be high due to low combustion chamber wall temperatures. (4) Throttling may be required to control HC emission at light load. (5) Plug erosion may be a problem even if sophisticated and expensive electrode materials are used. (6) Odour levels are high at light load. (7) Light load smoke is a problem on diesel fuel operation.

Figure 3.1 Layout of FM combustion system.

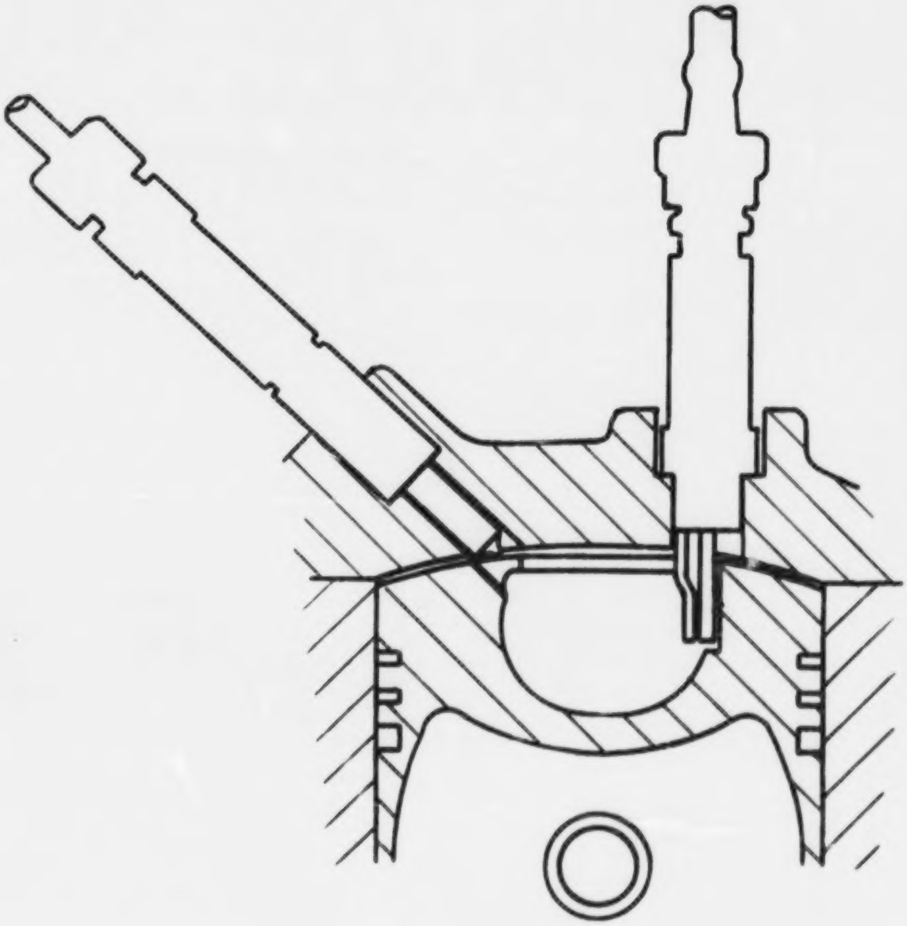
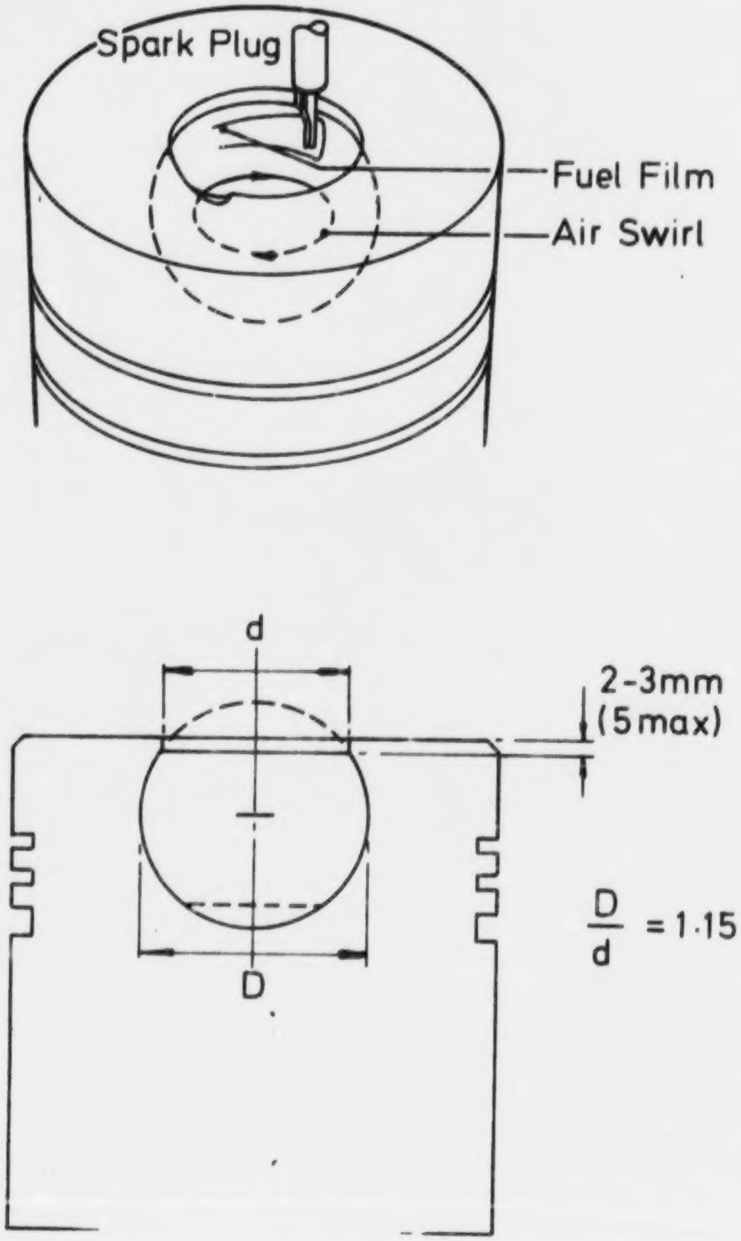


Figure 3.2 FM combustion bowl details.


Spark Plug
Fuel Film
Air Swirl
d
2-3mm
(5max)
D
$\frac{D}{d} = 1{\cdot}15$

Piston to Head clearance normally 1mm
Compression Ratio Variation of ±0·5
permissable due to piston tolerances.

Figure 3.3. Stratified charge study. Cylinder pressure diagrams for FM engine 105 x 109 mm. C.R. 15.8/1.

FUEL: GASOLINE RON 93. ENGINE SPEED: 2000 rev/min.

STATIC INJECTION AND IGNITION TIMING :-OPTIMUM FOR MIN. FUEL CONS.

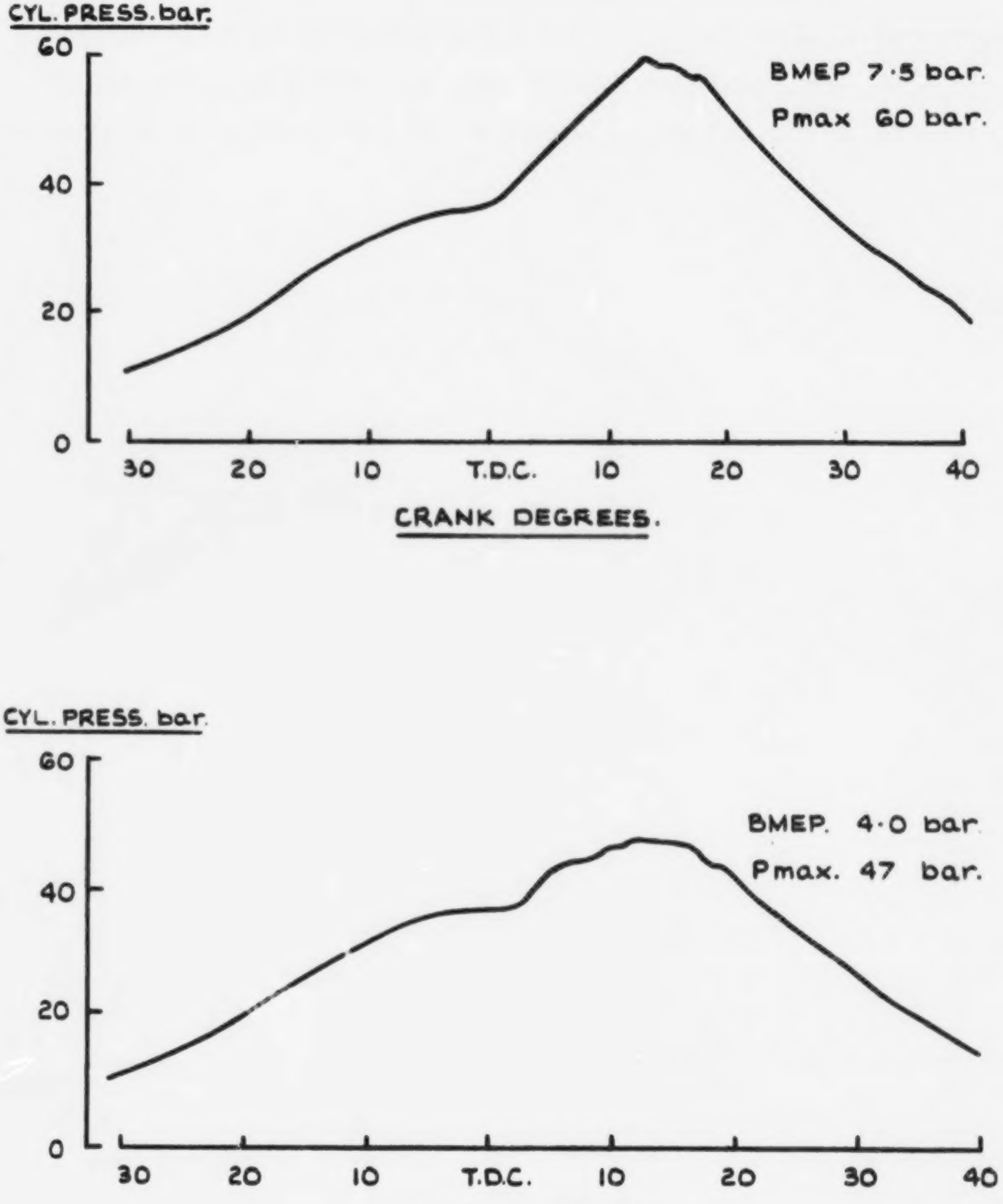


Figure 3.4. MAN-FM combustion system torque curves.

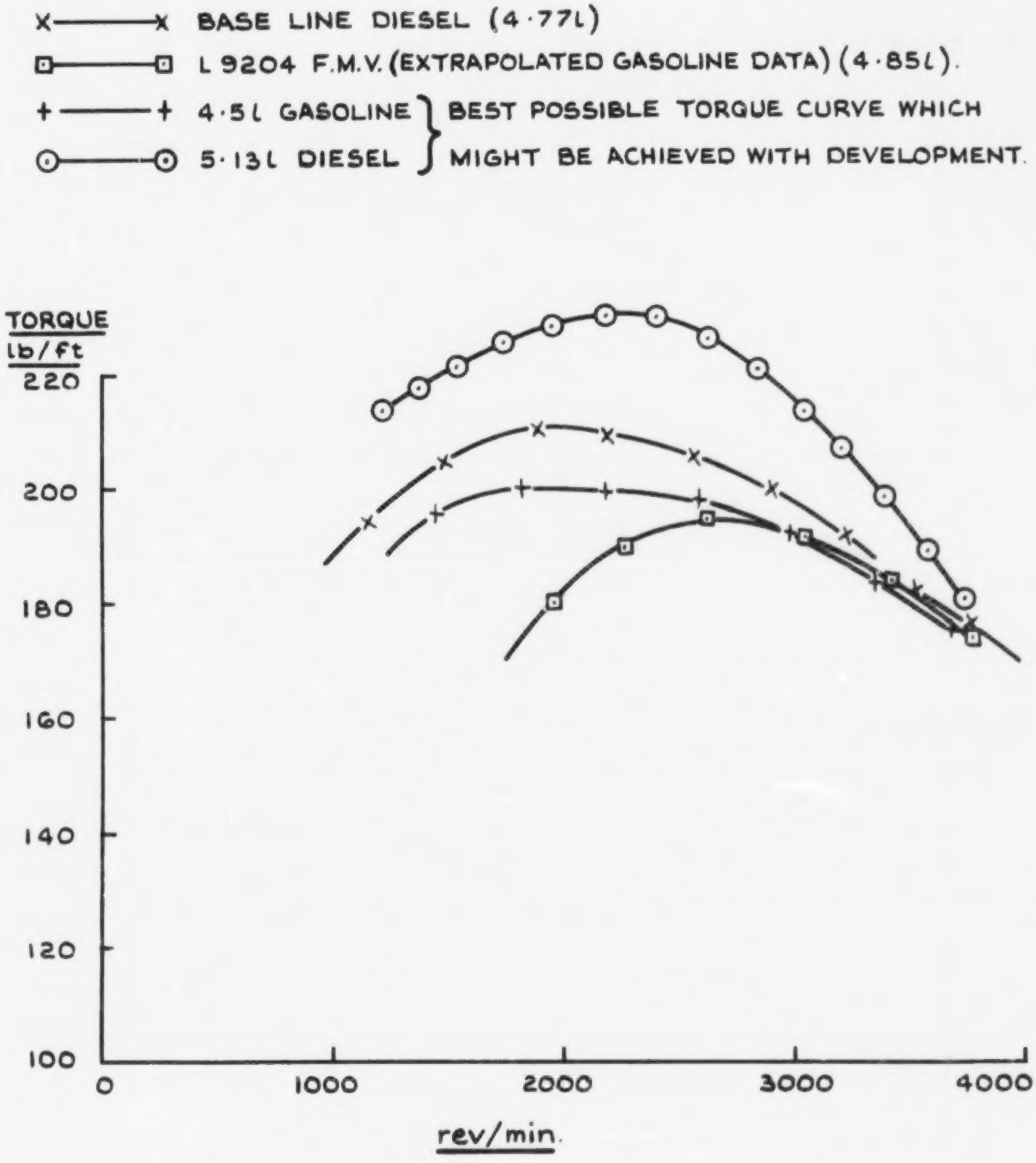


Figure 3-5. MAN-FM combustion system BMEP curves.

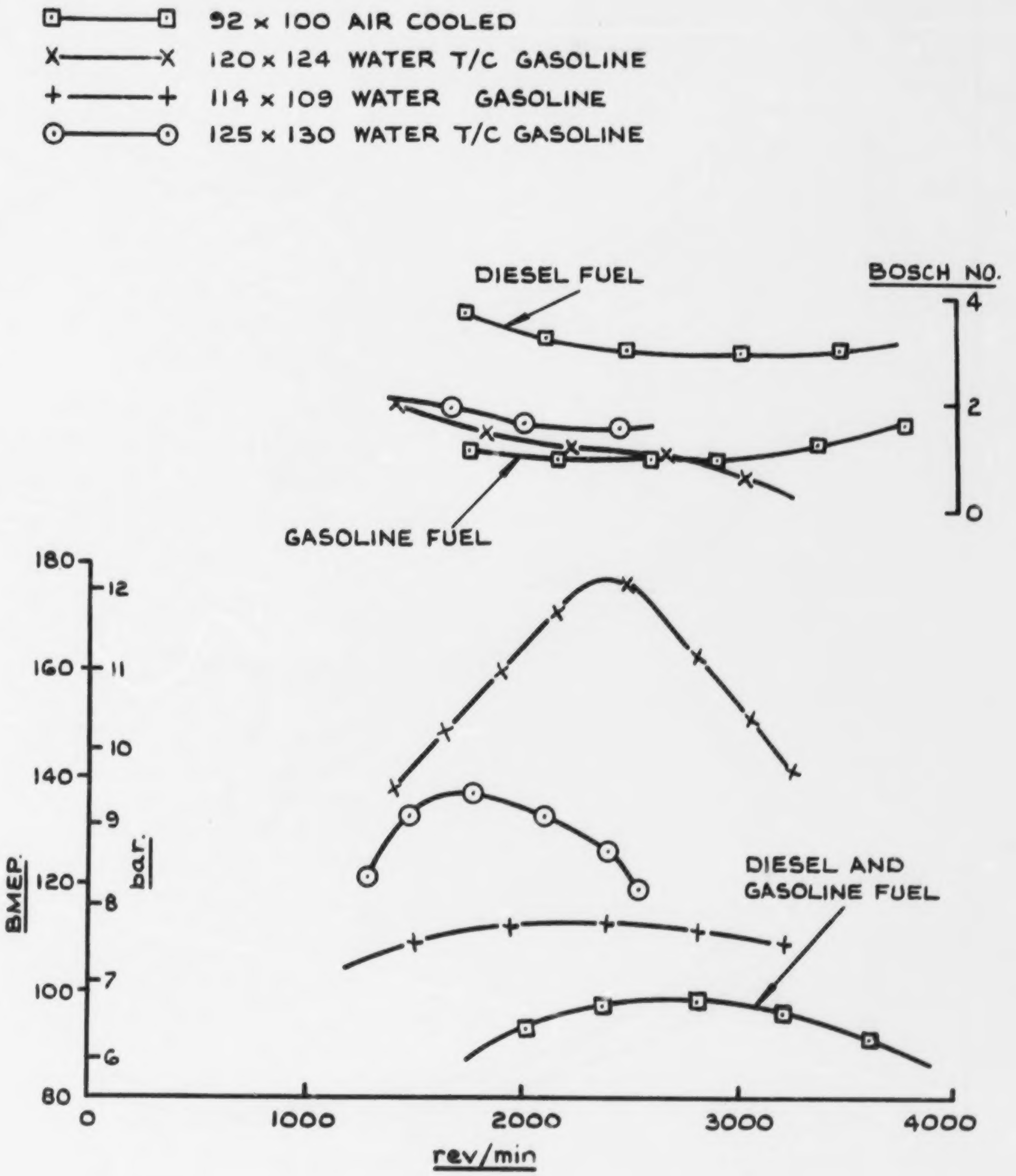


Figure 3.6. Part load fuel consumption comparison between baseline diesel and M.A.N. - FM.

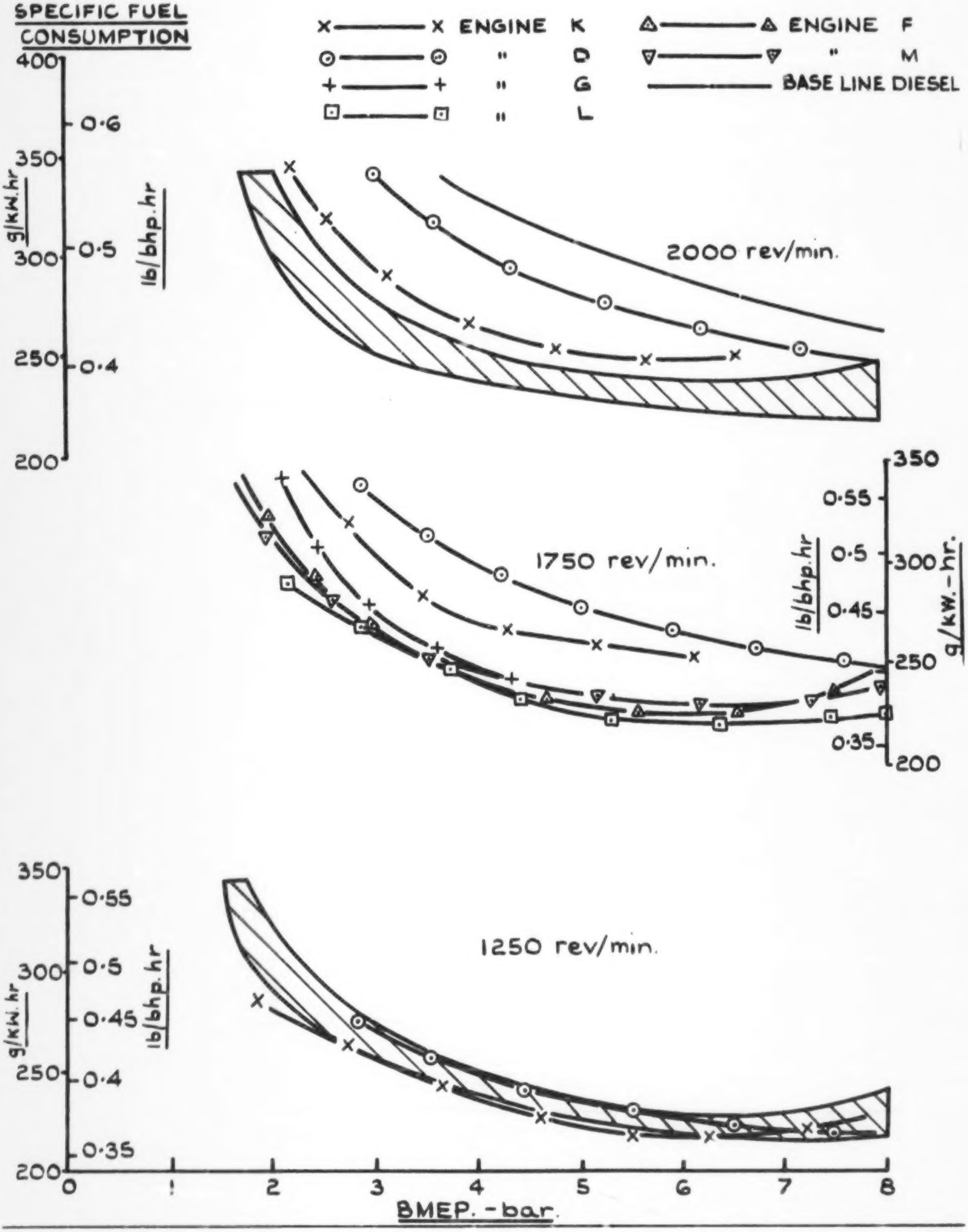


Figure 3.7. Part load fuel consumption comparison between baseline diesel and M.A.N. - FM.

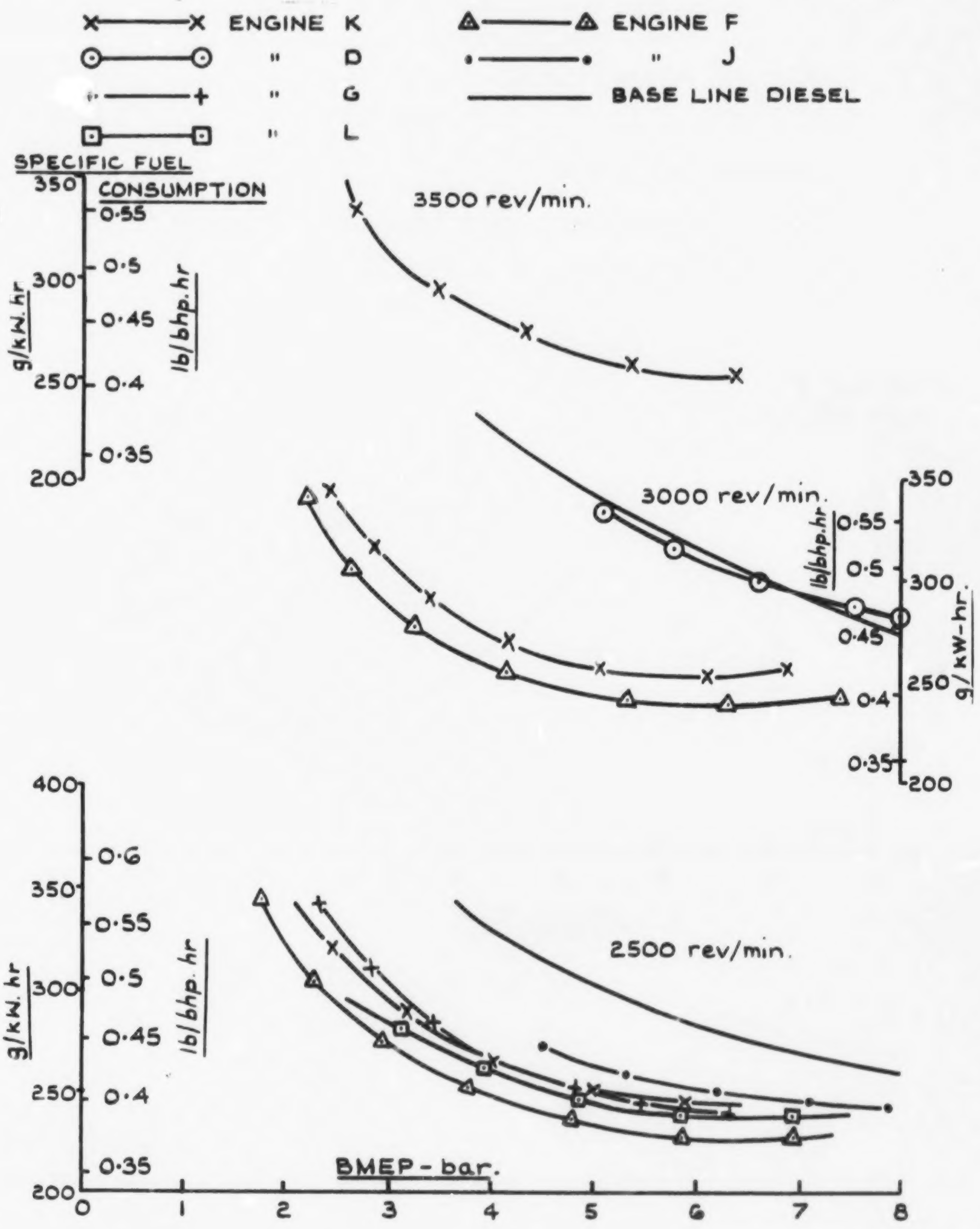


Figure 3.8. Part load fuel consumption comparison between baseline diesel and M.A.N. - FM.

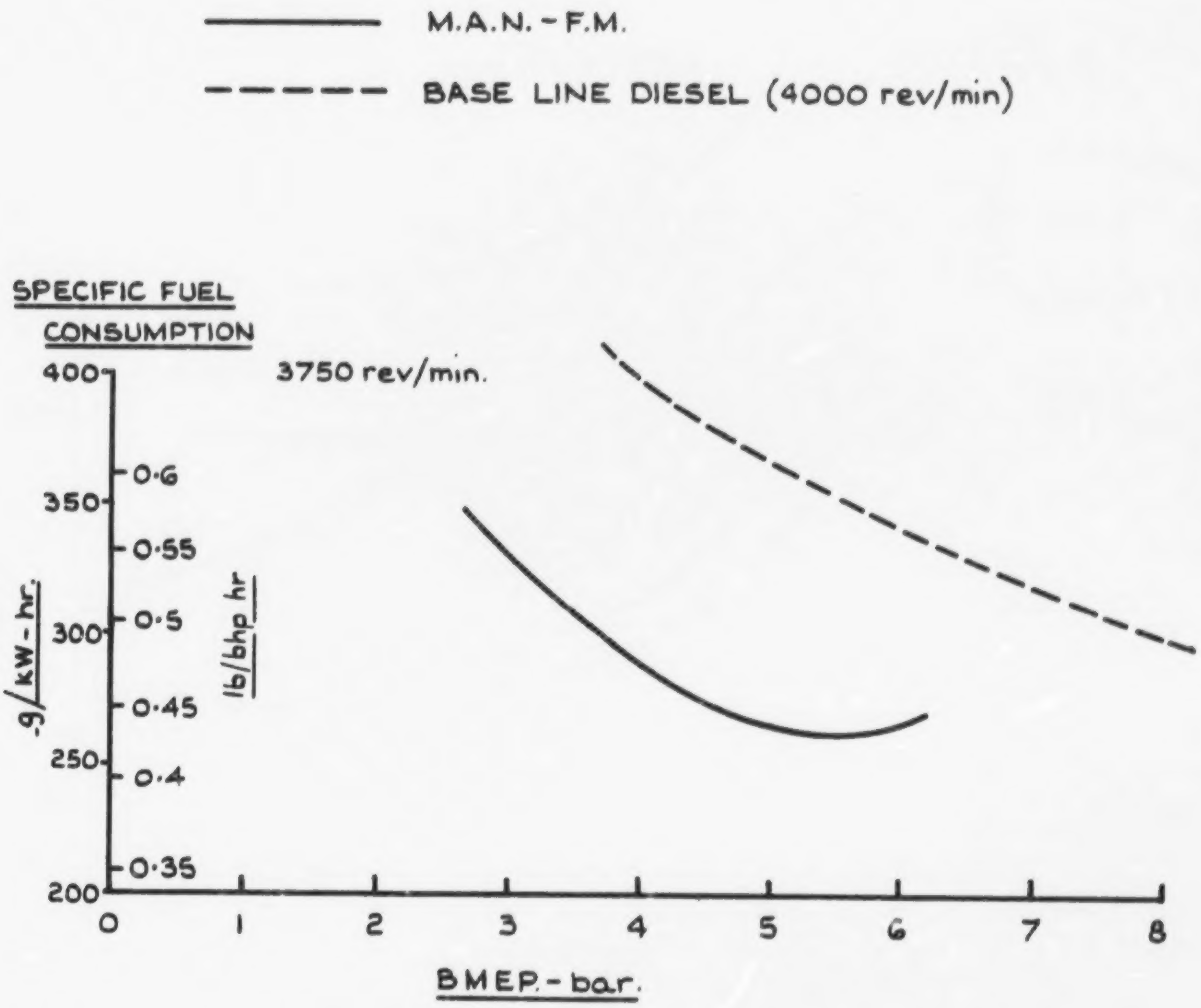


Figure 3.9. Emissions and fuel consumption of a "F.M." stratified charge engine extrapolated to the target horsepower and compared to the baseline diesel.

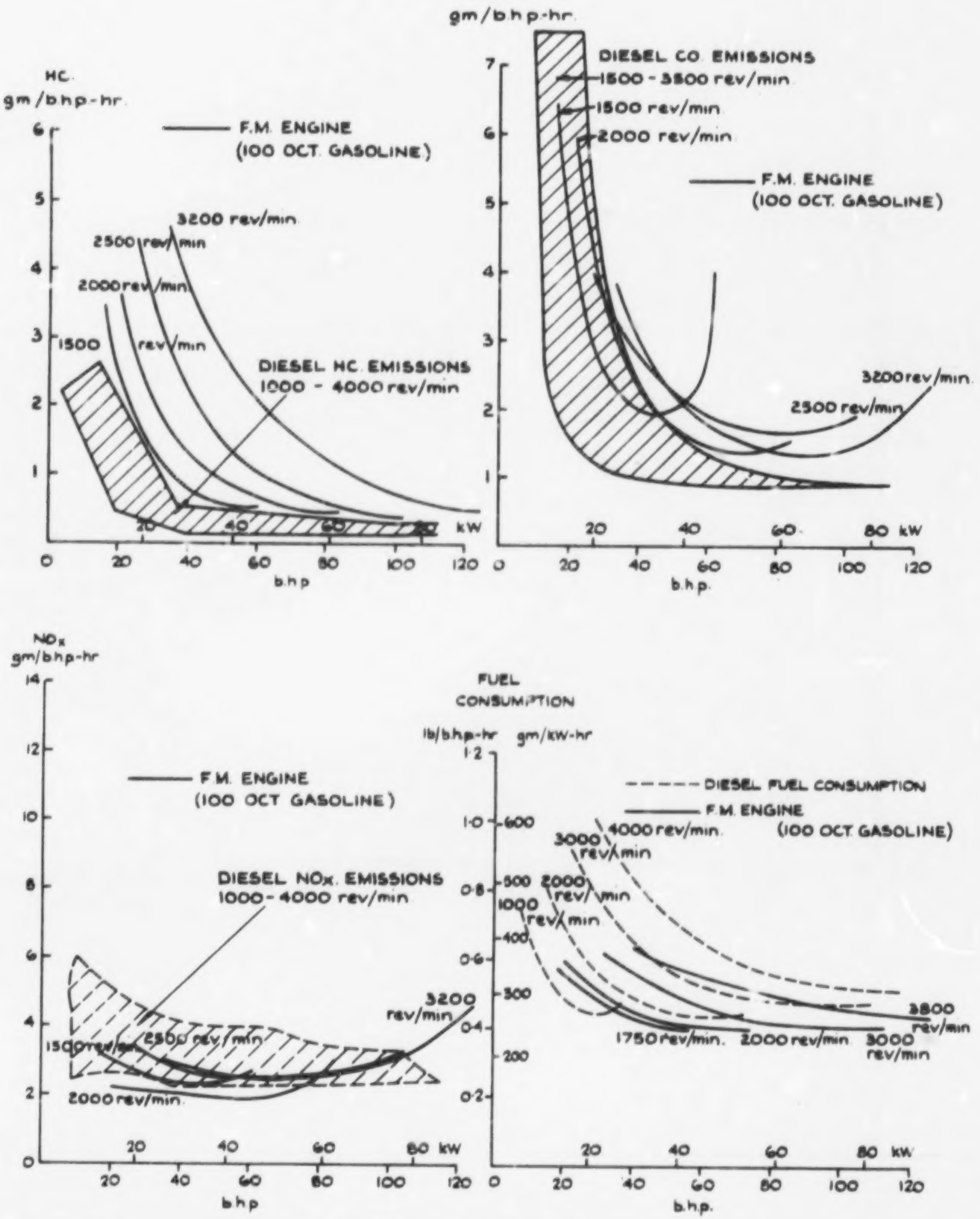


Figure 3.10. Comparison of friction of FM, diesel and gasoline engines.

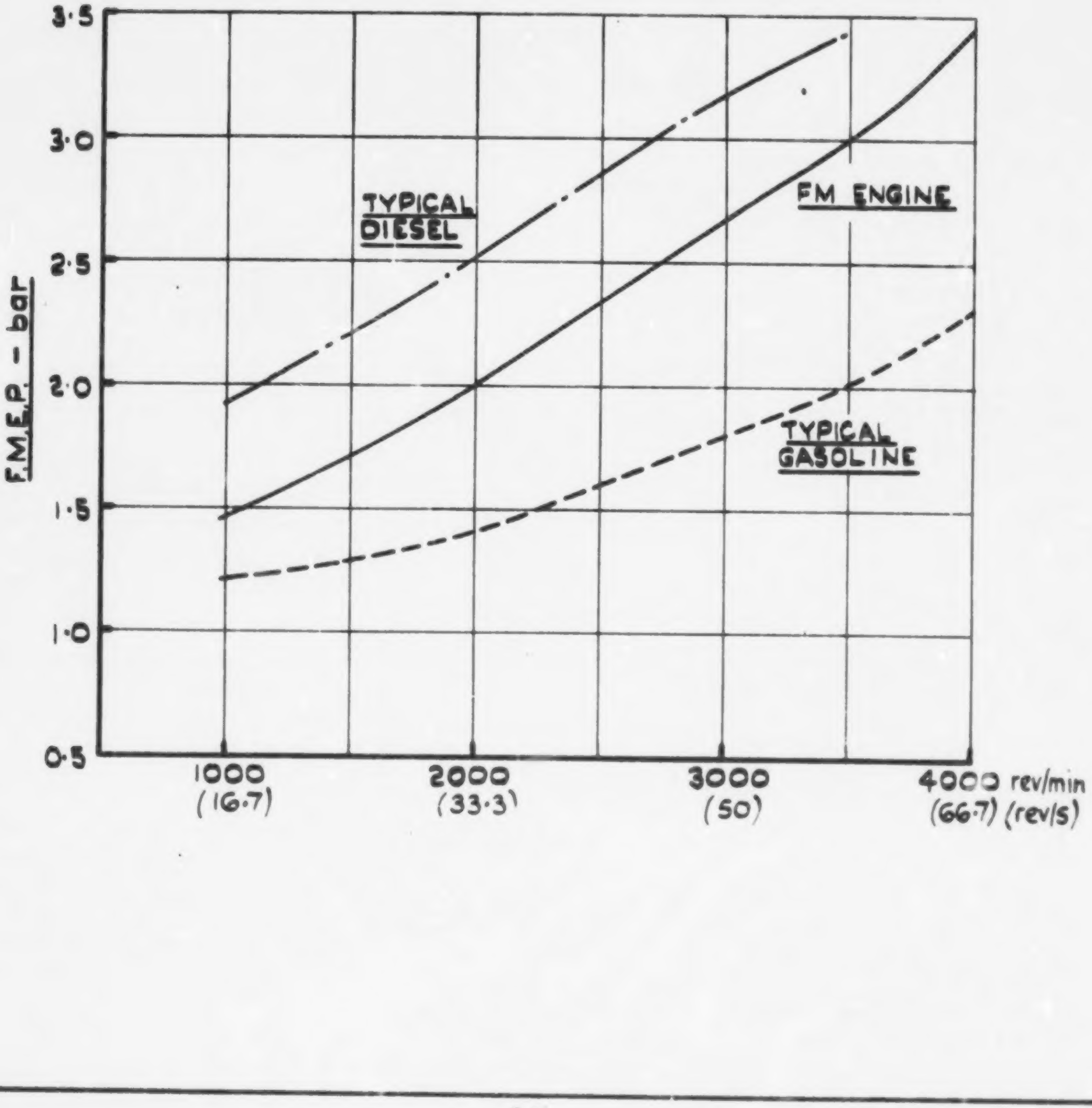
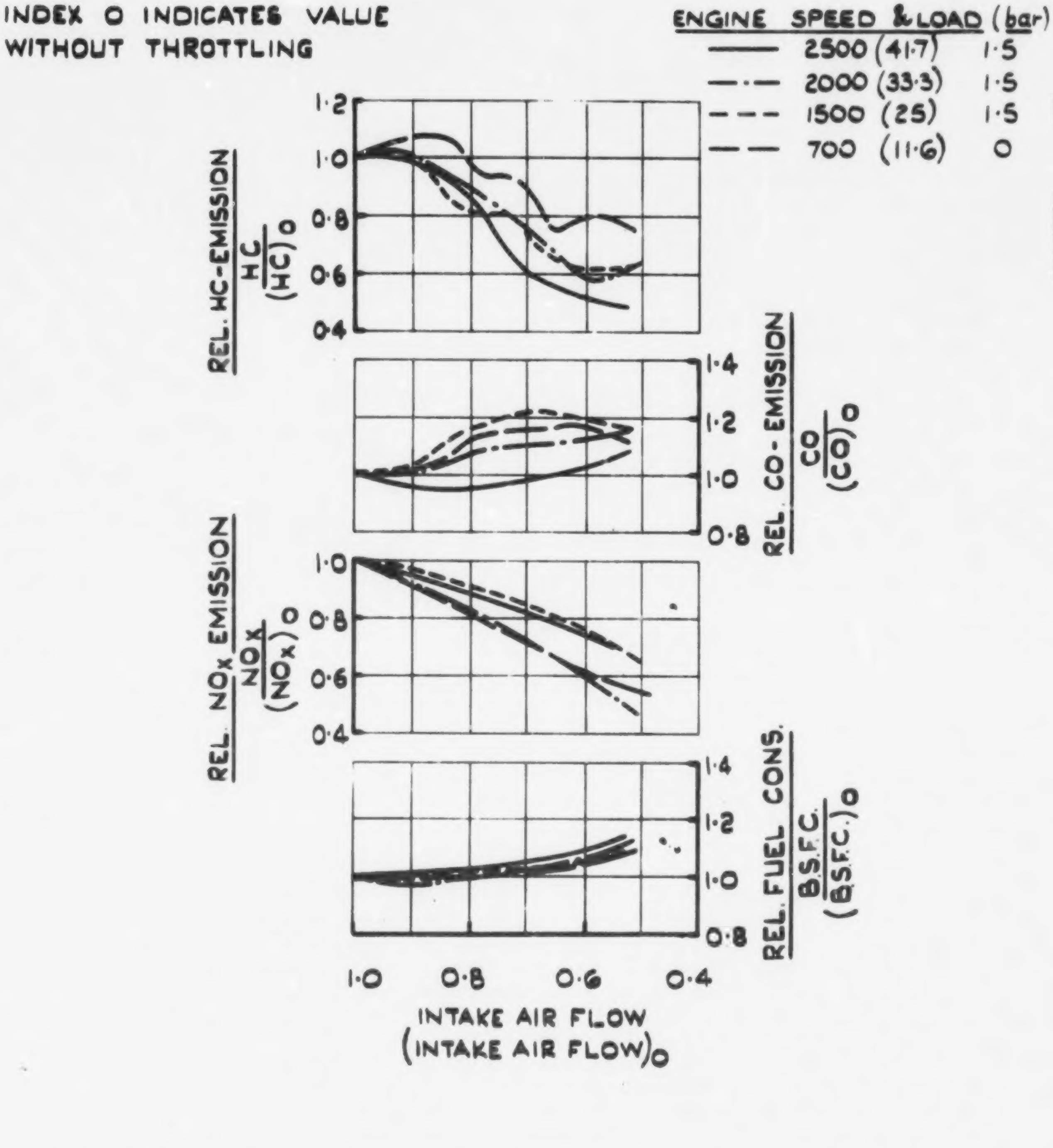


Figure 3.11. Effect of intake air throttling of exhaust emission and fuel consumption of FM combustion system.



CATEGORY IV

DIVIDED COMBUSTION CHAMBER WITH FUEL INJECTION INTO A PRECHAMBER

Introduction

The combustion systems in this category have not been separated into early, late and wall injection as with the direct injection types. There are no published results referring to injection controlled combustion in the manner of the TCCS engine. In addition, prechamber injection will probably always involve some wall wetting and some droplet penetration into the air, although the proportions of each of the effects will vary in different combustion systems.

The complexity of analysing the data for this category is further magnified by the large number of engine design and operating parameters, which can each alter the exhaust emission characteristics of the engine. Furthermore, in spite of the relatively large number of prechamber stratified charge engine investigations, data on the range of possible configurations is, by no means, complete. Therefore, a larger measure of engineering judgement had to be applied in this category to fill the gaps in the experimental data. As a consequence, the overall conclusions must contain a greater element of uncertainty than for the previous categories.

Topic Area (a) Combustion Chamber and Stratification Technique

The prechamber stratified charge engines can be conveniently separated into three general types depending on the proportion of the combustion chamber which is contained within the prechamber. The types are:

1. Prechambers which occupy less than 20% of the combustion chamber.
2. Those which occupy 20-40%.
3. Those which occupy more than 40%.

Type (1) resembles the conventional spark ignition engine, where the load is usually controlled by throttling the induced air, and the mixture strength remains approximately constant, independent of load. The prechamber is enriched by a separate fuel injection system, and burns quickly, throwing a jet of flame into the main chamber. Combustion is thus initiated at a number of points in the main combustion chamber, and a more reliable combustion is obtained. Therefore, this type of engine can generally operate at slightly leaner mixture strengths than for the conventional spark ignition engine. Engines of type (3) are usually unthrottled, since the prechamber fuelling alone can give a fairly wide torque range, provided the fuel within the prechamber is stratified at low loads. At higher loads, the prechamber is enriched over stoichiometric and the rich mixture is oxidised in the main chamber as it is expelled from the prechamber. Maximum torque is usually obtained by enriching the main chamber. Type (2) is a compromise between (1) and (3) and can be used in two modes. In the first mode fuel is only injected into the prechamber and the engine is not throttled. Prechamber fuelling alone will only provide sufficient power for idle and low load conditions. Medium and high loads are obtained by gradually enriching the main chamber with fuel. The alternative mode is to throttle the engine and maintain approximately constant air/fuel ratios in the prechamber and main chamber at all loads.

The various examples of prechamber stratified charge engines are described below, each being related to its appropriate type.

Freeman, type (1) ref. 4.2. This was a relatively crude example of a prechamber design as it was a simple chamber screwed into a CFR engine. The prechamber occupied 17% of the clearance volume, and the throat being very long and narrow, would reduce thermal efficiency, due to energy lost in heat transfer. The prechamber was cylindrical in shape, and there was no attempt to promote an orderly charge movement. Fuel was injected at the furthermost point from the throat and mixed with the air due to the turbulence generated by the throat, Fig. 4.1. Fuel was also added to the main chamber either by a carburettor or main chamber injection. Work was also carried out on an 8% prechamber. Much of the experimental work was carried out under unthrottled conditions, and the lean misfire air/fuel ratio was found to be around 50:1.

Porsche, type (1) ref. 4.3. Only a limited amount of information has been released on the Porsche engine, but it would appear that the prechamber occupies about 10% of the clearance volume, Fig. 4.2. The prechamber is of the swirl type, being cylindrical in form and arranged vertically. Fuel is injected from the top into the prechamber. A special characteristic of this engine is an anti-chamber for the spark plug which apparently reduces the turbulence level at the point of ignition and increases the probability of combustion initiation. This engine is throttled up to medium loads to maintain an overall air/fuel ratio of 25 and a prechamber air/fuel ratio of 10. At medium load the main chamber is gradually enriched, and at full load the prechamber fuelling is reduced to avoid misfire from overrich conditions at the plug. The misfire overall air/fuel ratio is 30:1. This combustion system has been studied on a single cylinder engine and a six cylinder version has been constructed and installed in a car. See Appendix 4.1.

Clawson, type (1) ref. 4.12. This again is a swirl prechamber although the form of the chamber is a sphere. The main chamber has been arranged so that the charge is all gathered below the exhaust valve and is easily accessible to the ignition flame from the prechamber. Fig. 4.3. The engine is throttled although the actual air/fuel ratio has not been specified. Fuel is introduced to the prechamber with an injector and to the induced air with either a petrol injection system or a carburettor. This combustion system has been applied to a Saab two-stroke and a L141 four-stroke engine in a Capri vehicle.

Broderson, type (2) ref. 4.5. This type of stratified charge engine has been studied for over 25 years and has gone through such an evolutionary process, that the present form does not resemble the original. The original Broderson engine was a novel arrangement, Fig. 4.4. The main intake valve was situated within the prechamber so that the complete air charge was drawn through the prechamber and throat during the induction stroke. The injector was arranged in the throat and the timing of the fuel injection governed the extent of stratification between the chambers. At light loads, injection commenced after the start of the compression stroke, so that the air being driven into the prechamber carried the fuel with it. Near full load the injection was advanced so that fuel was deposited at the throat and carried into the main cylinder. Injection continued into the compression stroke so that the prechamber was also enriched and so the fuel could be deployed in both chambers with only one fuel supply.

After trying various arrangements of the above concept, the noise problem associated with high rates of pressure rise in the large prechamber could not be overcome. The arrangement shown in Fig. 4.5 is the most recent adaptation of the Broderson concept and resembles most other fuel injected prechamber engines, except that the prechamber has a separate intake valve for scavenging. Prechamber sizes of 22-29% of the clearance volume have been tried on single cylinder engines. Misfire air/fuel ratios are above 120:1 if fuel is introduced to the prechamber alone. The remarkable air/fuel ratio tolerance of this type of engine is probably related to the efficient scavenging of the prechamber by the additional valve. Unscavenged prechambers have exhaust residuals remaining in the prechamber which dilute the compressed charge and affect ignitibility. Simple calculations at a compression ratio of 12 suggest that the ratio of residuals to air in an unscavenged prechamber is 2.2% at full load. However, as the load is reduced in an unthrottled engine, the temperature of the exhaust residuals decreases and the proportion of residuals rises to 5% at 1 bar BMEP. Therefore scavenging of the prechamber will certainly affect ignitibility.

Volkswagen, type (2) ref. 4.4. After a comprehensive study of scavenged and unscavenged small prechambers, VW began development of an unscavenged 30% swirl prechamber. The prechamber was of the spherical type and the throat was large with a smooth exit and entrance section to avoid compression throttling losses and to reduce turbulence, Fig. 4.6. The interrelation between turbulence, excessive combustion rates and engine noise was obviously appreciated. The original single cylinder engine could operate unthrottled at zero load,2000 rev/min at an air/fuel ratio of 45:1 without misfire. A four cylinder 1.6 litre version was constructed using a 20% prechamber, with fuel injection into both the prechamber and the inlet port and the engine was throttled to control HC emissions. The fuelling system was arranged such that a constant quantity of fuel was injected into each prechamber per cycle independent of engine load and speed. This is not ideal, although it simplifies the engine control. At full load, the prechamber injection enriched the induced charge by 30%. However, at light loads the prechamber injection enriched the charge by some 200% and this implies that the main chamber charge may have been leaner than strictly desirable at light loads.

Schlamann, type (2) ref. 4.9. This engine was a spark ignited version of a Ricardo Comet MK.III diesel engine cylinder head except that the prechamber occupied 25% of the clearance volume rather than 50%, Fig. 4.7. The experimental work was carried out on two single cylinder engines in the early 1950's and/or a multi-cylinder engine. The engines were unthrottled except at idle and fuel was introduced to the main chamber by a carburettor to control the load. The work was specifically directed towards knock resistance and low fuel consumption, and exhaust emissions were not recorded. The misfire air/fuel ratio of the engine was about 75:1.

Pischinger, type (3) ref. 4.6. The work by Pischinger on a spherical prechamber appears to have followed similar lines to the VW engine, Fig. 4.8. The prechamber occupied 42% of the clearance volume and most of the initial work was carried out under unthrottled conditions. The test engine was a single cylinder unit of a flat four. The development work was fairly comprehensive including different spark plug and injector arrangements and designs. Later the engine was throttled for HC control.

Huber, type (3) ref. 4.7. The Huber engine was also based on a spherical prechamber although the size was about 50% of the clearance volume, Fig. 4.9. Experiments were conducted with fuel introduced to the prechamber alone and with additional port injection and throttling. The single cylinder engine could be controlled to light loads on prechamber fuelling alone, by altering the injection timing. The air/fuel ratio at this condition was 55:1.

Newhall, type (2) ref. 4.8. This was also a single cylinder investigation with a large prechamber occupying 65% of the clearance volume, Fig. 4.10. The throat was short and no attempt was made to promote an orderly air movement. In this engine the spark plug was placed close to the prechamber throat under the assumption that the flame front would progress into the prechamber rather than advance toward the throat from a remote wall, and drive unburnt mixture into the main chamber. The misfire air/fuel ratio of this engine was 28:1.

Some of the published operating parameters of each engine are shown in Table 4.1 as follows:

Table 4.1. OPERATION OF ENGINES THAT USE PRECHAMBER INJECTION

Engine	Prechamber Volume / Clearance Volume	Engine Compression Ratio	Prechamber Injection Timing	Best Injection Pressures
Freeman	0.08-0.17	10	90° BTDC	-
Porsche	0.10	-	-	-
Clawson	0.15	11	-	-
Broderson (late)	0.22,0.29	8.8, 10	-	-
Volkswagen	0.2, 0.3	8.5	50-100°	25 bar
Schlamann	0.25	7.6, 11	170-90°	15-40 bar
Pischinger	0.42	9.5	60°	60 bar
Huber	0.5	9.0	45-60°	-
Newhall	0,65	8.0	100° BTDC	35 bar

Injection Equipment.

As shown in Table 1, the optimum fuel injection timings into the prechamber were all around the middle of the compression stroke to allow sufficient time for evaporation and mixture preparation. If the fuel is injected earlier, the mass of air in the prechamber is small and there is little charge movement and under these conditions fuel is deposited on the combustion chamber walls as the low density quiescent charge quickly becomes saturated with fuel vapour.

Atomisation of the fuel is obviously very important in these engines. It will be observed that the optimum injection pressures are low compared with diesel engine practice which is around 100 bar. The lower pressures of the stratified charge nozzles improve atomisation and reduce penetration of the droplets, some of which could impinge on the prechamber walls. In addition, the form of the spray is also important and wide spray angles are best. Pischinger found that an outward opening nozzle with a spray angle of 20° was much superior to more compact sprays. Newhall used an inward opening pintle nozzle with a 12° spray angle.

Schlamann developed an unusual pump concept for his engine to reduce the manufacturing costs compared to diesel injection equipment. The pump piston was operated by a cam which depressed a rubber seat and pressurised the fuel in the line, Fig. 4.11. In a prototype example the fuelling could be varied between 7 and 35 mm^3/inj.

Compression Ratio.

If the prechamber fuelling is supplemented by main chamber fuelling at full load then the engine cannot operate at compression ratios much higher than the existing gasoline engine as there will always be an end gas containing fuel which can detonate before flame arrival. There may be some latitude for increasing compression ratios which has not been realised in some of the engines above, as the main chamber mixture is usually leaner than a conventional engine. In the original Schlamann engine the compression ratio was 11:1 but the octane requirement was 102, so in the second engine the compression ratio was reduced to 7.6:1. This engine did not experience knock with a fuel of 75 RON provided the the fuel stratification was arranged so that the main chamber mixture was not richer than an air/fuel ratio of 17.5. The prechamber engines have high combustion rates which cause high noise levels. Some researchers have mistaken this noise as a form of detonation, and have therefore not reached the optimum compression ratio.

VW carried out octane number requirement tests on their multi-cylinder engine and found that the 8.5 compression ratio engine was completely satisfactory with regular grade gasoline.

Ignition Equipment.

A noticeable omission from all the references in this category are high energy ignition systems which were universally used for direct injection stratified charge engines. Most of the references do not mention ignition equipment and those which do apparently use the normal coil type. This may mean that high energy long duration sparks are not particularly beneficial for a prechamber stratified charge engine as is the case for the conventional petrol engine.

Two of the references in this category, i.e. Pischinger and Porsche, have observed that masking the spark gap from direct contact with the moving air improves the mixture ignitibility, smoke and HC emissions. Porsche mask the plug by placing it in a special ante-chamber within the prechamber and Pischinger turned the earth electrode of the plug so that the gap was masked from the direct air swirl.

Topic Area (b) Combustion Characteristics

The Achilles' heel of the prechamber stratified charge engine is the high noise level. This arises from two causes. In the first place, during the compression stroke a proportion of the gas is forced through the prechamber throat giving high turbulence levels in the prechamber. This, in turn, causes the mixture to burn quickly once it has been ignited by the spark. High burning rates and rates of pressure rise lead to an excessive noise level. In engines where fuel is also introduced into the main chamber, the fast combustion in the prechamber will accelerate the igniting torch coming from the throat. This will distribute the burning prechamber charge more fully throughout the main chamber, and also increase the combustion rate within the main chamber.

We shall now examine each of these causes of high noise level in more detail. The velocity of the air in the prechamber throat can be simply calculated as a function of the compression ratio and the ratio of prechamber volume to throat area. The resulting equation can be differentiated to find the maximum throat gas velocity which occurs in the region of 25-40^{o} BTDC (Appendix 4.3). The results of such an approach are shown in Fig. 4.12 for an engine speed of 1000 rev/min. The appropriate prechamber volume/throat area ratios are plotted from a number of experimental investigations. In addition, reference G16 has used hot wire anemometry to study charge movement in a conventional gasoline wedge combustion chamber immediately before combustion. These results are also shown on Fig. 4.12 at a speed of 1000 rev/min for a normal intake system and for a masked inlet valve. It is immediately apparent that the velocities, and therefore turbulence levels of prechamber engines are always above those of normal gasoline engines.

The velocity of the torch from the prechamber is more difficult to predict. However, a term can be derived to compare prechambers as shown in Appendix 4.2. This term is called the throat velocity factor, i.e.

$$\frac{FY\ (1 - F)}{A(CR)^{\gamma-1}}$$

The appropriate factors from a number of investigations are plotted in Fig. 4.13. Some Ricardo experimental programmes in this field are also included. The metamorphose engine incorporated a variable volume prechamber, ref. 4.11. The letter beside each data point refers to the comments of the investigator as to whether the noise levels were high, satisfactory or low. In some cases the authors of this survey exercise judgement on the noise level on the basis of the shape of the cylinder pressure diagram, and in one case (i.e. Huber) there were no facts of any sort available. In spite of these reservations a reasonable correlation exists between the noise level and the velocity factor. In one build of the Schlamann engine, the torch velocity was too low to completely ignite the weak mixture and the engine misfired at low loads.

Some examples of cylinder pressure diagrams from the Newhall, Porsche and Pischinger engines are shown in Fig. 4.14. In the Porsche engine the high throat velocities during compression cause very fast combustion which results in a large pressure difference between the two chambers. One of the engines in this category, i.e. Pischinger, was completely untypical of the others. Although the throat velocities were similar to many other prechamber engines the rates of pressure rises were very low, around 2 bar/deg. crank angle as in the conventional gasoline engine. This characteristic of the Pischinger engine cannot be explained.

The throat velocity factor also influences heat transfer in the prechamber throat. Freeman, who used high throat factors, had to take particular care with prechamber cooling. In Pischinger and VW engines, which had relatively low throat factors, fins on one side of the prechamber gave adequate cooling. Care must also be taken to avoid overcooling the prechamber as a number of investigators have shown that the wall temperature is a critical factor influencing the combustion process. This implies that evaporation of fuel from the prechamber walls is significant in terms of mixture preparation.

As with all the fuel injected stratified charge engines there is little combustion cyclic irregularity in prechamber engines. However, it would appear that for small prechambers the mixture in the prechamber must be appreciably richer than stoichiometric so that the torch which is emitted from the prechamber should contain hot partially oxidised components. If the mixture in the prechamber is leaner than stoichiometric, the combustion rate in the main chamber is reduced and cyclic irregulatity increases.

Topic Area (c) Performance

Table 4.2 below summarises the torque outputs of the various engines. Most of these results are estimated from single cylinder figures and refer to fuel addition to the main chamber as well as the prechamber.

Table 4.2. MAXIMUM SPECIFIC TORQUE OUTPUTS

Engine	Speed rev/min	Single Cylinder Result	Estimated Multi-Cylinder BMEP
Prechambers below 20%			
Freeman (induction port injection)	2000	9.68 bar IMEP	8.04 bar
(main chamber injection)	2000	9.34 bar IMEP	7.74 bar
Porsche	2000	7.19 bar BMEP	8.1 bar
Prechambers 20-40%			
Broderson (early type)	1200	8.25 bar IMEP	6.94 bar
(later type CR = 12)	1800	8.94 bar IMEP	7.54 bar
(later type CR = 11.2)	1200	9.27 bar IMEP	7.96 bar
Pischinger	2000		8.6 bar
VW	2000		7.9 bar *
Schlamann	1400	8.6 bar BMEP	9.2 bar
Prechambers greater than 40%			
Huber	2000	6.8 bar BMEP	7.5 bar
(fuel in prechamber only)	2000	5 bar BMEP	5.7 bar
Newhall (fuel in prechamber only)	1600	6.9 bar IMEP	5.6 bar

* Actual results from 4 cylinder VW PCI engine

The output of these engines is quite acceptable and little below that of modern U.S. gasoline practice. The particular results in the table which are low, can be explained. In the case of the Huber and Newhall engines with fuel addition to the prechamber alone, the air utilisation of these engines was apparently poor. In other words, the torch of burning gas coming from the prechamber was not mixing with the air in the main chamber and completing the fuel oxidation process. When fuel was added to the main chamber of the Huber engine, the specific torque output rose to the same level as the other engines.

The low torque output of the early Broderson engine was attributed to poor volumetric efficiency, i.e. the need to draw all the air through the prechamber orifice affected the engine breathing. No figures were available on the performance of the Clawson concept although the torque was said to be low. It is possible that the compact form of the main combustion chamber immediately below the small exhaust valve, masked the valve and affected the scavenging of residuals.

Smoke has been observed in the exhaust of some of the engines in this category, i.e. Pischinger and Porsche. In general, the maximum torque of the engines does not seem to be limited by smoke emissions as none of the references mention a smoke level when recording maximum torque. For engines where fuel exists in both chambers, maximum torque usually occurs at an overall air/fuel ratio of 12-13 as for the conventional gasoline engine.

The engine speed corresponding to maximum power is generally not known for most of the engines in this category. The exception is the four cylinder Volkswagen engine where the maximum power occurred at 4000 rev/min. The torque curve for the engines has only a limited torque back-up, Fig. 4.15. This, in fact, may be a normal characteristic for engines in this category. Since the indicated efficiency of prechamber engines is slightly below that of direct injection types, the maximum torque for a given volumetric efficiency is reduced. However, engine volumetric efficiency is not penalised at high engine speeds as there is no intake restrictions due to carburettors or swirl ports. Therefore, the engine torque does not fall off quickly at higher speeds, as do DI stratified charge engines which require intake swirl. The good high speed torque leads to a compact engine, if maximum power is the criteria for comparison.

A vehicle performance comparison between a standard Volkeswagen engine and the stratified charge version revealed no differences in acceleration performance. The calculated rear wheel torque during the CVS test also indicated that the 1.6 litre stratified charge engine should be able to follow the LA4 driving cycle in a 1000 kg car.

Topic Area (d) Fuel Consumption

For simplicity of control and optimum fuel economy a stratified charge engine should not be throttled. Prechamber stratified charge engines sometimes experience difficulty in meeting this requirement, particularly if the prechamber is small. Usually some load control is achieved by admitting fuel to the prechamber alone, but at some point in the load range fuel must also be added to the air in the main chamber. At combustion, the mixture is often too lean to be burnt effectively by the torch from the prechamber unless the velocity and mass flow of gas from the prechamber are high. This means that there is a certain minimum size of prechamber for unthrottled operation. An inspection of the literature suggests that this minimum size is in the region of 30-40%. The problem is illustrated particularly well in the "late" Broderson engines. Fig. 4.16 shows indicated thermal efficiency against fuel/air equivalence ratio for a number of engines. The general trend is for indicated thermal efficiency to rise slowly as the overall mixture strength becomes leaner. However, the Broderson engine has a dip in the curve corresponding to slow or incomplete burning when the mixture in the main chamber is lean. Schlamann identified the same problem. The effect usually presents itself as relatively poor fuel consumption, high HC emissions and some instability in engine operation.

The problem can be partially alleviated by combustion chamber design, i.e. a high throat velocity factor (e.g. Freeman) and good presentation of the air to the prechamber torch. In engines where the torch is directed at the piston, load stability is often poor. If most of the main chamber volume is arranged close to the prechamber throat, the results are usually better (Porsche & VW). The solution used by Schlamann was to introduce induction swirl and charge movement in the main chamber. In other words, mixture not ignited by the initial torch was swept by the main chamber swirl into the region immediately in front of the throat. However, the only complete solution for small prechambers is to partially throttle the engine so that an easily ignitible mixture of air and fuel exists in the main chamber at all loads. The Porsche engine was throttled to an overall air/fuel ratio of 25 at low loads and enriched beyond stoichiometric at full load. Pischinger and VW also adopted throttling at part loads with richer mixture strengths than Porsche. The Porsche engine had good part load economy, Fig. 4.18, while the Pischinger engine was similar to the diesel baseline, Fig. 4.19. The superiority of the Pischinger engine at higher speeds was probably due to lower frictional levels. The VW engine did not follow the expected trend and part load economy was relative poor, Fig. 4.20. An inspection of the published data suggests that the ignition timing of this engine was retarded at low loads to reduce HC emissions. In the CVS test fuel economy results from the stratified charge "Beetle" car covered a wide range, but none were better than the baseline for good conventional gasoline engines, Fig. 4.21. The worst fuel economy figures from the car probably correspond to the retarded timing conditions of the test bed figures when HC and NO emissions were at a minimum. Results from a Ford Capri fitted with a Clawson concept combustion chamber were no better than the baseline gasoline engine, Fig. 4.22. This trend is also indicated by the test bed results, recorded with EGR. Fig. 4.23.

The instability problem associated with the unthrottled engine with a small prechamber can usually be avoided with a large prechamber, i.e. prechamber size greater than 40% of the clearance volume. Huber achieved very low fuel consumption figures, particularly at part load, Fig. 4.24. Newhall's results with a 65% prechamber were much inferior. This was probably caused by the poor attention paid to the flow discharge characteristics of the torch as it entered the main chamber. The charge was directed against the piston crown. The graph of indicated thermal efficiency, Figs. 4.16 and 4.17, clearly suggest that very little post-oxidation was occurring within the main chamber. In the Huber engine, the throat was directed obliquely and the main chamber clearance volume was gathered below the prechamber throat.

It is interesting to observe that in all cases the full load fuel consumption of prechamber stratified charge engines is usually worse than gasoline or diesel engines. This can be attributed to two sources:

1. The compression ratio is lower than the diesel, therefore fuel consumption can be expected to be higher.

2. The prechamber torch causes a general agitation of the charge within the chamber which increases heat transfer. In addition, the surface to volume ratio of the prechamber engine is higher than for a conventional gasoline engine, and thus fuel consumption at full load can be expected to be worse than a conventional gasoline engine.

Topic Area (e) Emissions

Test Bed Results: Nitrogen Oxides.

The particularly attractive aspect of the engines in this category is very low levels of nitrogen oxides emissions. In particular, the levels can be a minimum at low speeds and loads which will result in very low mass levels in city driving. The reason for this unusual result is that the fuelling to small prechambers is usually arranged at a constant level independent of the engine load. Since the "small prechamber" engines are throttled at low loads, the mixture in the prechamber is usually very rich, often bordering on the rich misfire regime. In addition, the mixture in the main chamber is lean. Therefore, the combustion occurs in comparatively rich and lean regimes which both produce low NO_x levels. At higher loads, the amount of stratification is usually reduced, so that the NO_x levels increase. This characteristic is not displayed by the unthrottled large prechamber engines as the prechamber itself is stratified at low loads. Nevertheless, the NO_x levels are still remarkably low.

There are no published results on the effect of EGR on most of these engines, except for the Clawson concept. In this case 20% EGR was used at idle, reducing to 5% at 4.8 bar BMEP. The results indicate that the low prechamber NO_x levels can be further reduced with EGR. Since part load economy of the prechamber stratified charge engine is already good, applying EGR to these concepts might prove to be the most promising method of achieving 0.4 g/mile NO_x with the minimum fuel economy sacrificed. While some direct injection stratified charge engines have better part load economy than prechamber engines, a considerable economy/emission trade off is required to meet 0.4 g/mile NO_x.

Hydrocarbons.

A comparison of Figs. 4.18 to 4.20 with the PROCO and TCCS results in categories I and II illustrates that the HC levels of these engines are relatively good, although still not sufficiently low to achieve 0.41 g/mile HC without exhaust after-treatment. The exception is the Clawson concept where HC levels are exceptionally high, Fig. 4.23. The HC levels of the Porsche and Pischinger engines are similar to current U.S. gasoline engines, as the homogeneous mixture in the main chamber still leaves a quench layer in the combustion chamber which will be released as hydrocarbons. The advantage of the prechamber stratified charge engine would appear to be in burning weaker mixtures of 18-25 air/fuel ratio without the sudden increase in hydrocarbons experienced by conventional gasoline engines at these mixture strengths.

The test results on HC emissions from the Volkswagen engine were uncommonly low, and did not correlate with the relatively poor emission results from the vehicle. The instrument used to measure HC was not specified in the literature, and indirect evidence suggests that it was an infra-red type. This instrument would under-estimate the HC emissions by at least 50%. HC may also have been low due to retarded ignition timings.

Large prechamber engines should give exceptionally low HC emissions since fuel mixture and quench layers are absent from the main chamber. Any quench layer in the prechamber should not be released during the exhaust stroke. The Newhall engine confirmed this theory, and achieved HC levels down to 0.5 g/ihp h. Unpublished Ricardo work on an unthrottled 50% prechamber has achieved similar results ranging from 0.3 to 1.2 g/ihp h over the load range. Unthrottled Huber engine results were less impressive, Fig. 4.24, perhaps indicating that spark plug and injector position and detailed combustion chamber design exerts a large influence on hydrocarbon emissions. In an unsatisfactory design, the combustion may drive unburnt mixture into the main chamber before it can be burnt, where it will be quenched and released as hydrocarbon emissions. There is some disagreement in the published literature as to what constitutes good design practice for prechamber engines. Porsche found that hydrocarbon emissions were improved with a large prechamber throat while VW observed the opposite effect.

Throttling of prechamber engines was found to be advantageous in reducing HC emissions at part load. This applied to all engines in this category, both with and without fuel in the main chamber.

CO Emissions.

The CO emissions of all the engines in this category followed the expected trends with very low levels at part load conditions, and rising at full load.

Vehicle Results.

The VW and Clawson stratified charge engines are the only examples in this category which have been applied to a vehicle. A range of CVS results for each of these engines are shown in Figs. 4.21 and 4.22. The minimum NO_x levels are very low and come close to the secondary emission target. The relatively high CO levels of the VW are due to high concentrations at full load. The high vehicle HC results of the VW engine are very much greater than the test bed figures, even by taking account of any difference in instrumentation. It may be that the engine has a control problem in maintaining a constant air/fuel ratio under transient conditions. The injection system was specially constructed for this stratified charge engine. The results from the Clawson engine, Fig. 4.22 were recorded with a catalyst in the exhaust system.

Smoke.

There is very little published information on smoke emissions from these engines. They may, or may not, have significant smoke emissions depending on the degree of richness of the prechamber. Only Pischinger has recorded smoke levels, which are around Bosch No. 2 at 6 bar BMEP. However, one cannot deduce whether this level would increase or decrease as the main chamber inlet charge was further enriched. If the prechamber fuel flow was reduced then the smoke should also decrease. The smoke at any load is a function of the prechamber air/fuel ratio, which may not necessarily become richer as load increases.

Topic Area (f) Multi-fuel Capability

The references in this category do not claim multi-fuel capability in the accepted sense. This is to be expected as it is difficult to use heavy fuels in carburettors or with port injection. Even if this were possible, the fuel would not mix properly with the air in the cylinder.

The Porsche engine has operated on methanol and has exhibited the same general trends as observed with methanol in conventional gasoline engines, i.e. improved thermal efficiency and a reduction in NO_x. The effects of methanol on HC emissions are more difficult to measure as the normal FID instrument does not respond to partially oxidised hydrocarbons.

Topic Area (g) Engine Components

The prechamber is the heart of the combustion system and the temperature of the walls affects the combustion process. Porsche found that the optimum wall temperature was in the range of 150-200°C. Some investigators have found it necessary to cool the prechamber with a separate coolant supply (e.g. Freeman) while others have found that short cooling fins on one side of the chamber were adequate. Heat transfer to the prechamber walls is governed by prechamber swirl level and throat velocity during compression and combustion.

The published literature on this category does not refer to ignition systems other than the conventional coil and contact breaker arrangement. This would probably be satisfactory for the small throttled prechamber engines although a long duration high energy system might be preferable for unthrottled prechambers since stratification exists within the prechamber at low loads. Spark timing is usually governed by engine speed and may include a small correction for load. The small throttled prechamber engines are generally fairly tolerant of incorrect setting in ignition timing, but the larger unthrottled prechambers can misfire if the ignition settings stray by more than about 10 crank angle degrees.

The injection system requirements depend on the size of prechamber and the use or absence of throttling. The one feature common to all the prechamber stratified charge engines is that the maximum fuel pressures are modest (i.e. up to 40 bar) in comparison with diesel practice and this simplifies component manufacture. In addition, the tolerance on injection timing is wider than for diesels or some direct injection stratified charge engines (e.g. PROCO). Thus, less expensive components can be used, e.g. distributor pumps rather than in-line pumps.

Unthrottled Engines.

A single injection system is satisfactory for these engines and load is simply controlled by the quantity of fuel injected. If the maximum power is to be augmented by fuel addition to the main chamber, then a simple continuous flow port injection with the flow controlled by needle valves will suffice. The main complication of these engines is that the injection timing must be modulated over a wide range depending on both speed and load. The total timing variation required may be up to 90 crank degrees.

Throttled Engines (without fuel addition to main chamber).

The injection pump fuelling control must now be interlinked with a throttle to give a constant prechamber air/fuel ratio. Therefore, the control problems are identical to those of port injection in conventional gasoline engines and the fuelling must be governed by manifold depression and engine speed. However, this is partially compensated by a reduction in the range of injection timing with load.

Throttled Engines (with fuel addition to main chamber).

The fuelling control problems of these engines are more complex than any other stratified charge engine. Two injection systems must be integrated together to supply the specified prechamber and main chamber air/fuel ratios at all engine conditions. The researchers in this field have simplified the approach such that a fixed quantity of fuel is injected into the prechamber per cycle irrespective of engine loads and speeds. This fixed quantity of fuel must be small to avoid the prechamber becoming too rich at low loads. Volkswagen developed a special double injector pump for their four cylinder engine, which contained six plungers. Four plungers supplied a constant quantity of fuel to the prechambers while the other two plungers supplied the cylinder ports, each injecting into two ports simultaneously. The following problems were experienced:

1. Since the fuel quantity delivered to the prechamber is very small, around 2 mm^3/inj, difficulties arise in maintaining equal flow in different cylinders and consistent performance between engines in production.

2. The air and fuel flow in the main induction system must be maintained in a fixed ratio and thus the air flow must be sensed by engine speed and manifold depression or throttle angle. The amount of fuel required must then be computed either electronically or mechanically with a 3-dimensional cam. The fuel flow control system must also react quickly to avoid lean misfires.

An eight cylinder engine based on this combustion concept would be expensive to manufacture due to this complex control system and the need for two fuel injectors in each cylinder.

Topic Area (i) Mathematical Models

The amount of effort which has been expended on mathematical models of prechamber engines is very limited, due to the complex computing procedures and restrictive assumptions involved in creating a model. Pischinger has produced a very simple zero dimensional model of general stratified charge combustion to show that the NO_x levels are considerably reduced compared to homogeneous combustion. However, the applicability of this model to solving other problems is limited. Mathematical models of prechamber engines are discussed further under Category V.

Topic Area (j) Engine Size and Weight

The 1.6 litre flat four VW engine was adapted from the standard gasoline model. The inclusion of a prechamber and injector in the cylinder head did not materially affect engine size and the only additional engine component was the injection pump. This probably added about 11 kg to the total engine weight. Generally, combustion systems in this category could be integrated into the structures of conventional gasoline engines without affecting the overall dimensions. Since the compression ratios and peak cylinder pressures are only slightly greater than normal, increased bearing sizes are not needed. If noise were a problem area, shielding might be required.

The extra weight of the injection system will be governed by the type of system used. Mechanical injection systems such as that used by VW are relatively heavy, but if an electronic system could be adapted for this type of engine the extra weight would be minimal.

Topic Area (k) Noise

This is usually the most important problem area for this category of stratified charge engine, although there has been one exception where high noise levels were not observed (i.e. Pischinger). The large unthrottled prechambers are the worst in this respect. The reasons for the high noise levels have been explained under topic area (b). Usually the noise is a function of engine load and is highest at full load, although one investigator (Varde) has observed maximum noise in the mid-load range. The noise usually has a large high frequency component which is harsh and annoying to the human ear, and is difficult to remove other than by complete engine shielding with acoustically damped panels.

Topic Area (l) Durability and Starting

The published literature does not contain information on the durability aspect of any of the engines in this category. From experience with other stratified charge engines one might expect problems due to carbon deposits on the injector which would affect the spray characteristics. This is usually a problem with medium pressure in-cylinder injection, as the fuel pressures are not high enough to scour carbon deposits from the injector. In addition, thermal loading of the throat may eventually cause cracking, particularly in engines using high throat velocity factors.

Cold starting of the VW engine was not straightforward as in the direct injection stratified charge engines. This may be due to the influence of prechamber wall temperature on the fuel evaporation. VW found that an extra injector had to be incorporated behind the throttle blade for cold starts. The residence time of the fuel between throttle blade and cylinder allowed sufficient time for fuel evaporation. The extra air for the cold start was supplied by a slide valve, and both the slide valve and cold start injector were progressively closed by temperature sensors in the engine coolant, once the engine was running. During decelerations, a decel valve supplied sufficient air to the engine to support combustion. If fuel cut-off were applied during deceleration, the prechamber cooled and combustion and emissions were affected.

Turbocharging.

The literature does not contain information on turbocharging of any of the engines in this category. Indeed, the potential gains from turbocharging are relatively small if mixture is introduced into the main chamber. The compression ratio would probably have to be reduced to avoid the onset of knock. If mixture were not admitted to the main chamber, then the full load torque would be reduced and the turbocharging would only compensate for the lack of power.

Rotary Configurations.

In theory, the prechamber combustion principle should also be applicable to rotary engines and Curtiss-Wright have carried out exploratory tests. However, the results do not appear to have been encouraging. There is always a high concentration of residuals remaining in the prechamber as the prechamber is never exposed to exhaust or intake pressure for very long and this would probably affect the combustion process.

Late Injection Engines

In all the engines in this category the fuel injection has occurred before combustion commenced. It is possible to partly control combustion by fuel injection during the combustion event, in the same manner as TCCS. The advantage would be that the noise problem of early injection would be reduced, and the potential advantage of prechamber engines for low HC and NO emissions will be retained.

Other Arrangements

Some prechamber engines have been uncovered by this survey, which have different concepts to those considered in the main text.

The Ricardo metamorphose engine had a variable volume prechamber where the prechamber piston was interlinked with the fuel injection pump. Fig. 4.28. At low loads, the prechamber volume was reduced, thus maintaining a richer and more easily ignitable mixture in the prechamber. The compression ratio also increased as the load was reduced, in an attempt to achieve better fuel economy. The engine operated reasonably well, although the design parameters (i.e. prechamber throat length and orientation) were known to be far from optimum. Good fuel consumption was demonstrated at low loads (e.g. 3 bar IMEP, CR 14:1, A/F 60, 30% prechamber) but at full load the fuel consumption was poor (e.g. 8 bar IMEP, CR 8.5, A/F 16, 62% prechamber). As usual, light load hydrocarbon emissions were the main problem area.

An unusual prechamber injection system design has been patented by Aerobonner Ltd., to avoid the cost of a fuel injection pump. Fig. 4.27. A pumpless injector delivers the fuel by using a differential area piston operated by the gas compression pressure to force the fuel into the prechamber. This is adequate for prechambers using early fuel injection, but it would not function if the delivery pressures were set above 13 bar. The prechamber fuelling may either be fixed, independent of load and speed, or else varied, using a cam to control the movement of the differential area piston. The concept of the patent envisages a small prechamber with fuel also introduced to the induced air.

Application to Motorcycles

In comparison to direct injection stratified charge engines, the problems associated with prechamber arrangements are even more acute. Not only is the injection equipment even more expensive, but water cooling may be unavoidable. Thermal loading of the metal in the region of the prechamber throat is always high, and it would be difficult to arrange adequate cooling by fins. In addition, the prechamber size becomes very small even in larger motorcycle engines of 250 cc cylinder size.

Conclusions

1. Nine types of stratified charge engines have been identified in this category and they have been sub-divided into three groups, depending on the ratio of the prechamber volume to the overall combustion chamber volume. The three ratios considered were below 20%, between 20 and 40% and over 40%.

These engines are usually operated in a throttled mode although the larger prechambers can be operated unthrottled over a relatively wide load range. Fuel is always introduced to the prechamber by an injector, and fuel may also be added to the main induced charge, with either a port injector or a carburettor.

The correct design of the prechamber, throat and main combustion chambers is ratner uncertain, and small changes in design can significantly affect emissions and economy. This requires further fluid dynamic studies, using perhaps combustion photography to determine the optimum prechamber design.

2. The most important problem area for this category of stratified charge engine is noise. This is due to the high turbulence introduced by a proportion of the air being forced through the prechamber throat, and in turn this high turbulence causes the mixture to burn very quickly. This problem appears to be fundamental to the use of a prechamber.

3. If fuel is introduced to the main chamber a reasonable maximum specific torque of around 8 bar can be expected from the engine. If the proportion of main chamber and prechamber fuelling is arranged correctly, this maximum torque will not be accompanied by smoke in the exhaust gases.

4. Most engines with small prechambers must be throttled to avoid an unstable region in the power spectrum. However, even these engines can operate at significantly leaner mixtures than the conventional gasoline engine. Test bed fuel consumption figures from some engines have been good, rivalling those of the diesel. However, the one vehicle fuel economy result from the VW engines was disappointing, and was lower than the baseline for good gasoline engines.

5. Nitrogen oxide emissions from prechamber engines are exceptionally low, and vehicle results below 1 g/mile without EGR can be achieved. There are no vehicle results referring to the target of 0.4 g/mile NO_x but this should be possible with EGR. Indeed, the economy sacrifice in meeting 0.4 g/mile NO_x would probably be less severe than for direct injection engines.

   HC emissions are similar to those of the conventional gasoline engine. However, if fuel is not introduced to the main chamber of the engine, HC emissions may be reduced. Prechamber engines are unlikely to meet 0.41 g/mile HC without a catalyst, although the baseline HC levels might be lower than for the direct injection engines.

6. These engines do not seem to have a multi-fuel capability.

7. The operating pressure of the prechamber injection systems are around 40 bar. Since an additional port injection system is usually required, the fuelling system becomes very complex. In addition, the fuel flow must be controlled by the air flow, and metered correctly to each cylinder. Further components are required for cold starting as these engines have trouble starting on prechamber fuelling alone if the prechamber walls are cold. Therefore, the cost of such an injection system would be high. An electronic version of the system would be slightly less expensive than the mechanical version used by VW.

8. The literature survey did not uncover a mathematical model of a prechamber engine and there is a distinct need for such a multi-dimensional model to improve our understanding of the processes in these engines.

9. These stratified charge processes can be readily applied to the existing gasoline engine structure with little change in engine size and weight.

10. There is no information on the durability of these engines, although any problems which do arise in the future should be easily solved with some development.

11. Turbocharging or rotary configurations of prechamber stratified charge engines do not look particularly attractive.

12. A prechamber engine based on late injection might show greater potential than the early injection types reviewed in this category.

APPENDIX 4.1

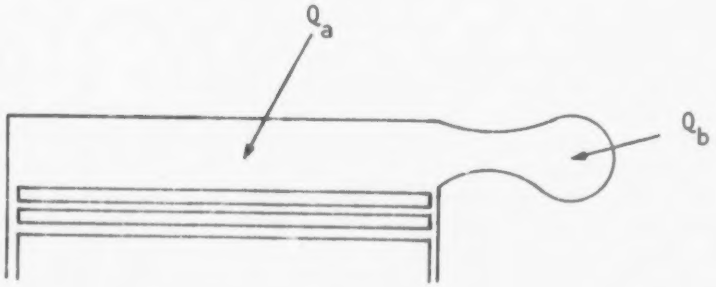
Recent Results from the Six Cylinder Porsche Stratified Charge Engine

After the text of this category had been written, new results were received from Porsche. These referred to a six cylinder version of the stratified charge engine which had been installed in a car. The test bed emission and indicated specific fuel consumption curves referred to a speed of 3000 rev/min. These results were translated into brake terms, and are shown in Figs. 4.25 and 4.26. The maximum specific torque was high (10.6 bar IMEP), exhaust emissions were low, and the fuel consumption was remarkably low. Vehicle results confirmed this low emission potential and some tests were within the 0.4 g/mile NO_x level. The HC levels exceeded 0.41 g/mile, which was surprising in view of the low test bed figures. These results were obtained without any form of after-treatment device, and presumably without EGR. Fuel consumption over the CVS test was not recorded.

These results are certainly the most attractive which have been revealed by this survey. However, Porsche have released no detailed information on this engine and judgement on the potential of the system must be deferred until further data becomes available.

APPENDIX 4.2

Derivation of Formula for Throat Velocity during Combustion in a Pre-chamber Stratified Charge Engine



Consider a very simple model, making the following assumptions:

1) During combustion the piston is stationary

2) There is a homogeneous mixture of air and fuel in the pre-chamber and only air in the main chamber.

3) The pre-chamber burns homogeneously, i.e. all the charge burns together at a linear rate over a time t_o. As the charge burns, the burning mixture expands through the pre-chamber throat.

4) The burning charge from the pre-chamber mixes completely with the main chamber air and the partially burnt fuel continues to burn at the same rate.

5) Pressure drop across the throat is zero.

6) Heat transfer is zero.

7) Perfect gas laws apply.

Notation

Q = heat addition
M = total mass of air
K = calorific value of fuel
F = ratio of pre-chamber volume to clearance volume
A = area of throat
t_o = time for combustion
ϕ = fuel air ratio in pre-chamber at beginning of combustion
T = temperature (a in main chamber, b in pre-chamber)
T_2 = initial temperature at start of combustion
C_V = specific heat at constant volume
Y = combustion chamber total clearance volume
CR = compression ratio
V = velocity of gas in throat
T_1 = ambient temperature

M_a = instantaneous mass of air in main chamber
M_b = instantaneous mass of air in pre-chamber

At any instant after combustion

$$Q_b = \phi\, m_b\, K \frac{t}{t_o}$$

$$Q_a = \phi\, (FM - m_b)\, K \frac{t}{t_o}$$

and

$$Q_a = m_a\, C_V\, (T_a - T_2)$$

$$Q_b = m_b\, C_V\, (T_b - T_2)$$

The equation of state can be written $\dfrac{T_b}{T_a} = \dfrac{F}{1 - F} \cdot \dfrac{M - m_b}{m_b}$

These five equations give

$$\frac{m_b}{M} = \frac{F\left(F\phi\, \frac{K}{C_V} \cdot \frac{t}{t_o} + T_2\right)}{\phi\, \frac{K}{C_V} \cdot \frac{t}{t_o} + T_2} \qquad \text{..........(i)}$$

$$T_2 = T_1 \ (CR)^{\gamma - 1}$$

The density of the gas at any instant is $\dfrac{m_b}{V_b} = \dfrac{m_b}{FY}$

The mass outflow from the pre-chamber $= \dfrac{dm_b}{dt}$

Volume outflow from the pre-chamber $= \dfrac{FY}{m_b} \cdot \dfrac{dm_b}{dt}$

$$\therefore \quad V = \frac{FY}{A} \cdot \frac{1}{m_b} \cdot \frac{dm_b}{dt} \qquad \text{..........(ii)}$$

Differentiating (i) and amalgamating with (ii) gives

$$V = \frac{FY}{A} \quad (1 - F) \ \frac{L}{(FLt + 1)\ (Lt + 1)} \quad \text{where } L = \phi \ \frac{K}{C_V} \ \frac{1}{t_o} \cdot \frac{1}{CR^{\gamma - 1}} \cdot \frac{1}{T_1}$$

By inspection we see that this formula has a maximum at t = 0

$\therefore$ Maximum velocity in throat

$$V = \frac{FY}{A} \cdot (1 - F) . \ \frac{\phi K}{C_V} \cdot \frac{t}{t_o} \cdot \frac{1}{(CR)^{\gamma - 1}} \cdot \frac{1}{T_1}$$

Since we are seeking a rough guide to compare pre-chamber engines we can assume that ϕ is the same for all engines at full load. (This in fact, is not true, but the error is small compared to other assumptions which have been made.) Since K, C_V and T_1 will be the same for all engines, a simple expression is generated for comparing throat velocities.

$$V \ \alpha \ \frac{FY}{A} \cdot \frac{(1 - F)}{CR^{\gamma - 1}} \cdot \frac{1}{t_o}$$

Usually, the time for combustion is not known, so for expediency the throat velocity factor is assumed to be

$$\frac{FY}{A} \cdot \frac{(1 - F)}{CR^{\gamma - 1}}$$

APPENDIX 4.3

Derivation of Formula for Throat Velocity during Compression Stroke

Assumptions:

1) Sinusoidal piston motion.

2) No pressure drop across the throat.

Adopting the same notation as Appendix (1) and adding

SV = swept volume
N = engine speed
V = instantaneous volume in cylinder
α = crank angle
t = time

At any instant during the compression stroke

$$V = FY + \frac{SV}{2} \cdot (1 + \cos \alpha)$$

Volume flow through pre-chamber throat $= \frac{FY}{V} \cdot \frac{dV}{dt}$

$$\alpha = \frac{\pi Nt}{30}$$

Velocity through pre-chamber throat $= \frac{1}{A} \cdot \frac{1}{V} \cdot \frac{dV}{dt}$

$$= \frac{FY}{A} \cdot \frac{N}{60} \cdot \frac{\sin \alpha}{\frac{1}{CR - 1} + \frac{1}{2} + \frac{1}{2} \cos \alpha}$$

By differentiating, we can show that the maximum velocity in the throat

$$= \frac{FY}{A} \cdot \frac{\frac{\pi N}{30} \cdot \sin \alpha_m}{\frac{2}{CR - 1} + 1 + \cos \alpha_m}$$

$$\text{where } \alpha_m = \cos^{-1} \cdot \frac{1}{\frac{2}{CR - 1} + 1}$$

This equation has been drawn in fig. 4.12 for values of pre-chamber volume divided by throat area $\left(\frac{FY}{A}\right)$ for various compression ratios. Some points from pre-chamber engine investigations at Ricardo have also been inserted.

Figure 4.1. Freeman.

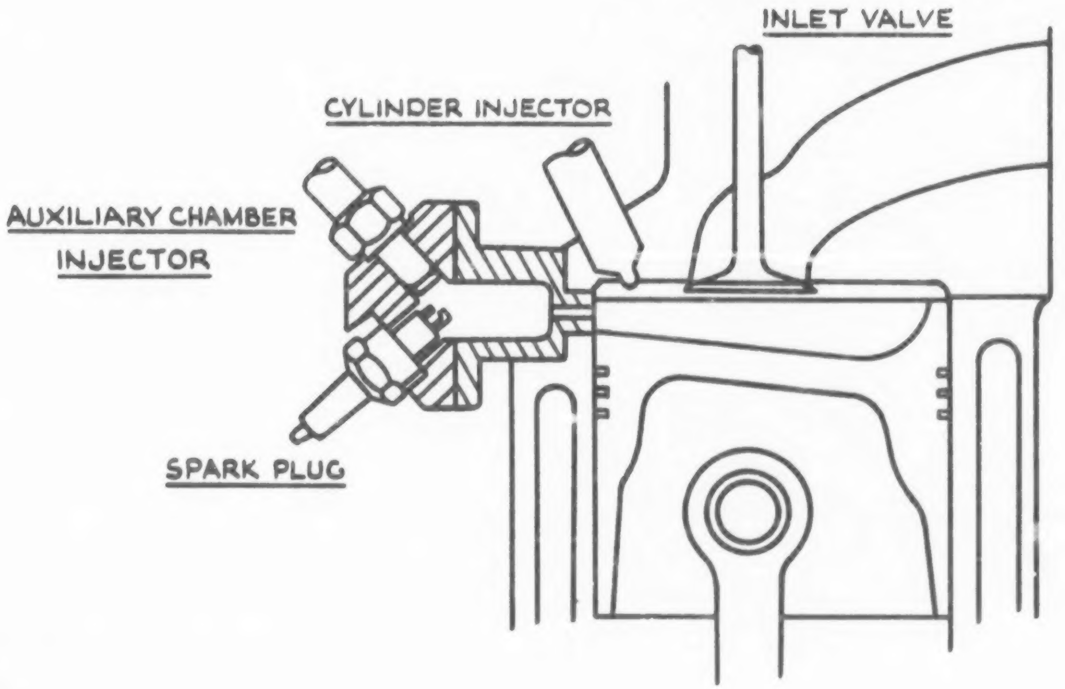

INLET VALVE
CYLINDER INJECTOR
AUXILIARY CHAMBER
INJECTOR
SPARK PLUG

Figure 4.2. Porsche.

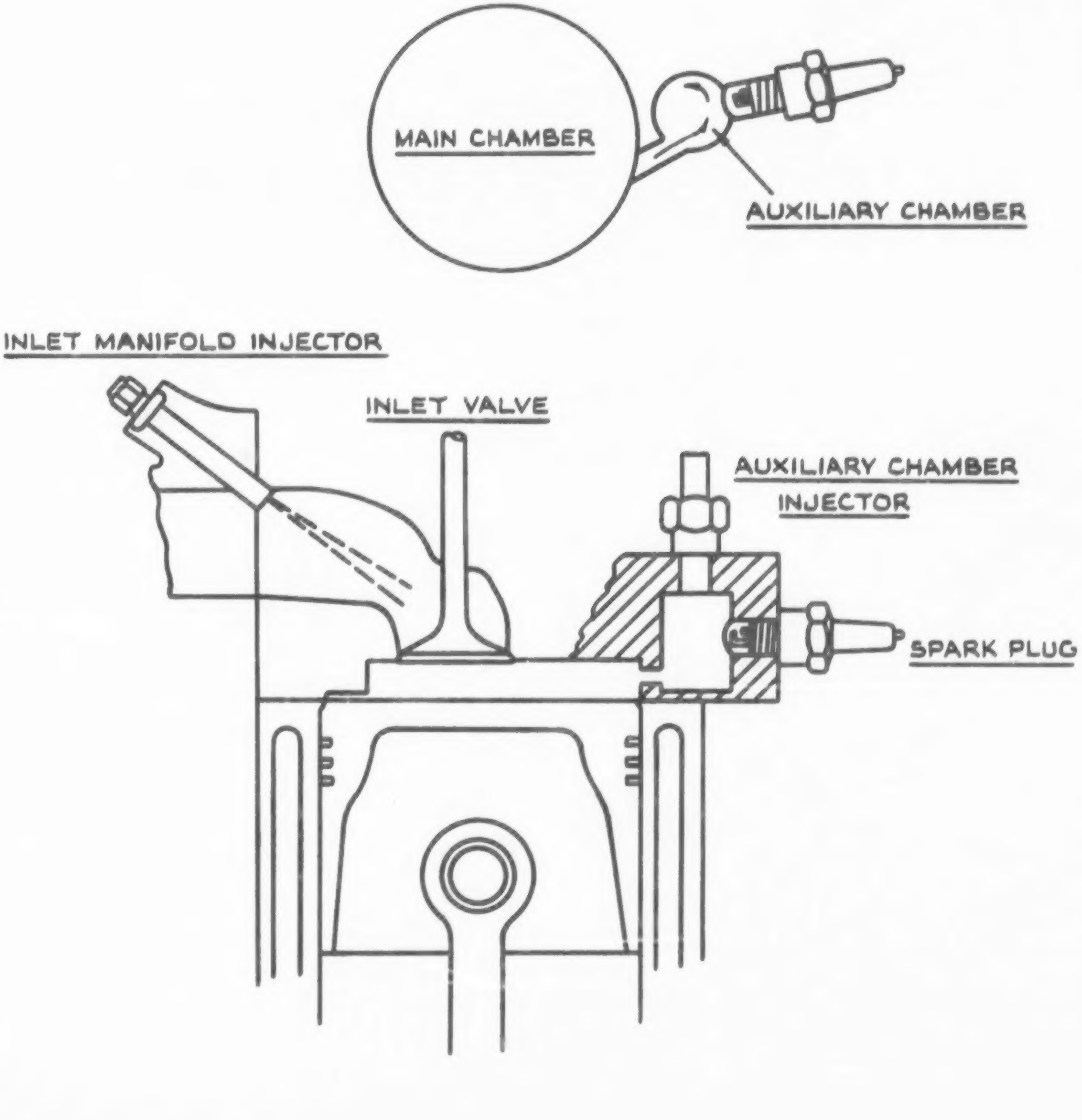

MAIN CHAMBER
AUXILIARY CHAMBER
INLET MANIFOLD INJECTOR
INLET VALVE
AUXILIARY CHAMBER INJECTOR
SPARK PLUG

Figure 4.3. Clawson.

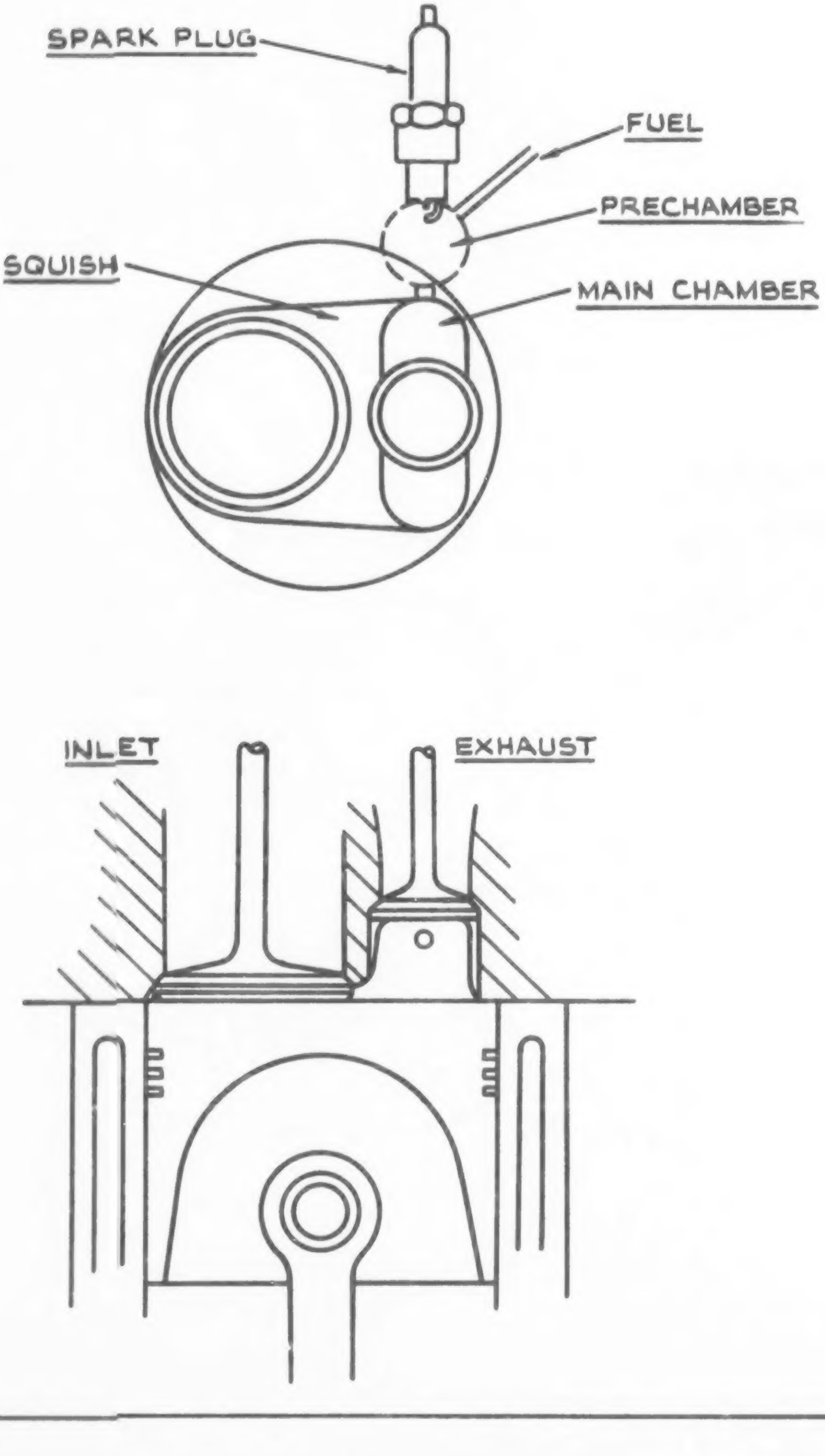

SPARK PLUG
FUEL
PRECHAMBER
SQUISH
MAIN CHAMBER
INLET
EXHAUST

Figure 4.4. Broderson (early).

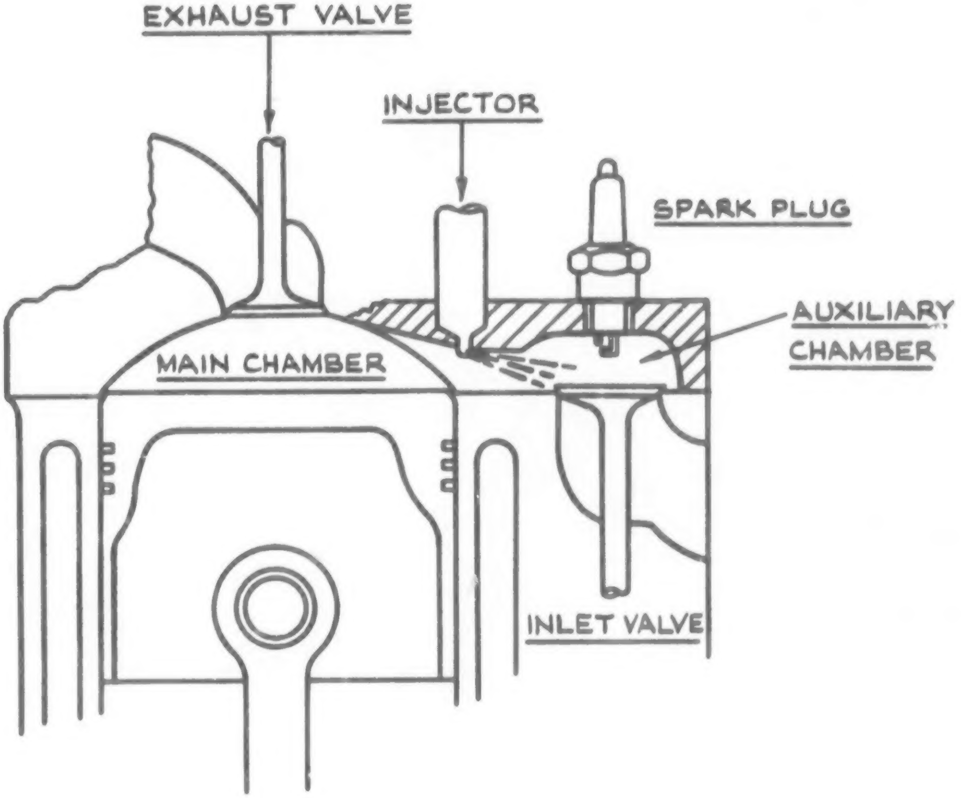

EXHAUST VALVE
INJECTOR
SPARK PLUG
AUXILIARY CHAMBER
MAIN CHAMBER
INLET VALVE

Figure 4.5. Broderson (late).

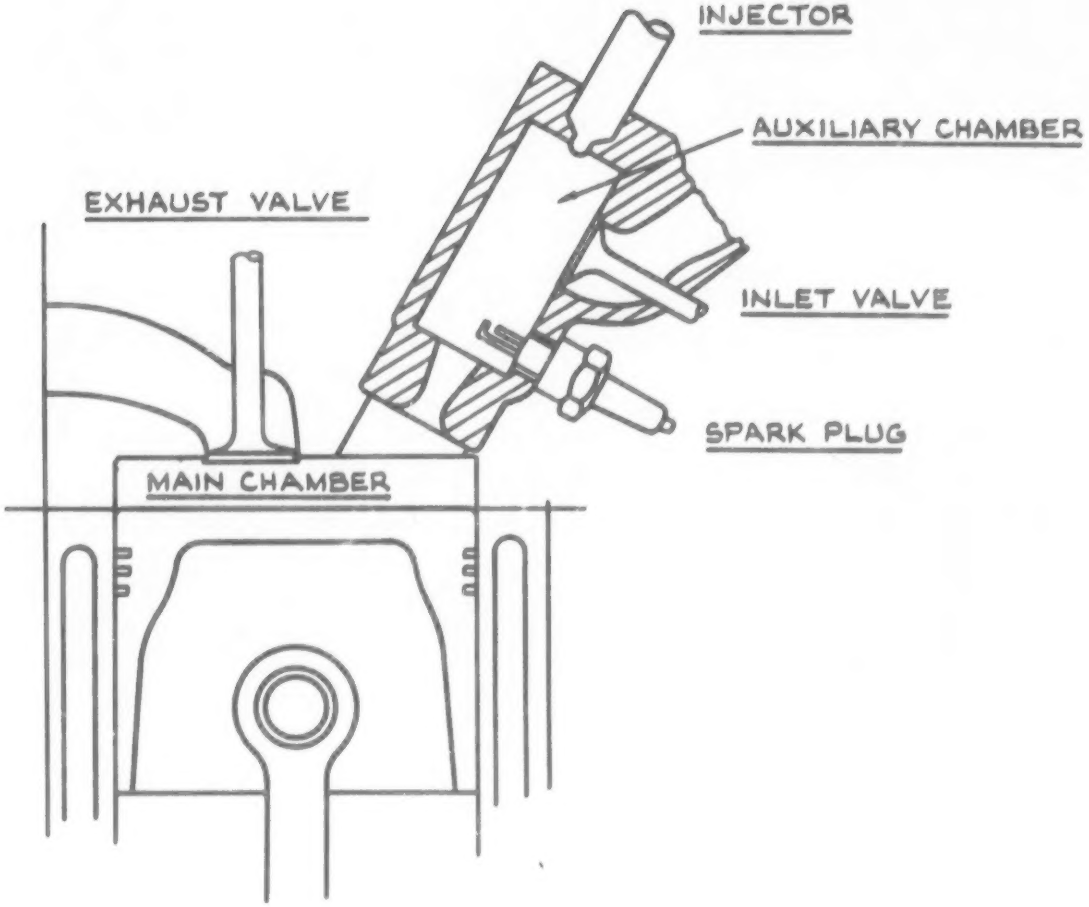

INJECTOR
AUXILIARY CHAMBER
EXHAUST VALVE
INLET VALVE
SPARK PLUG
MAIN CHAMBER

Figure 4.6. VW.

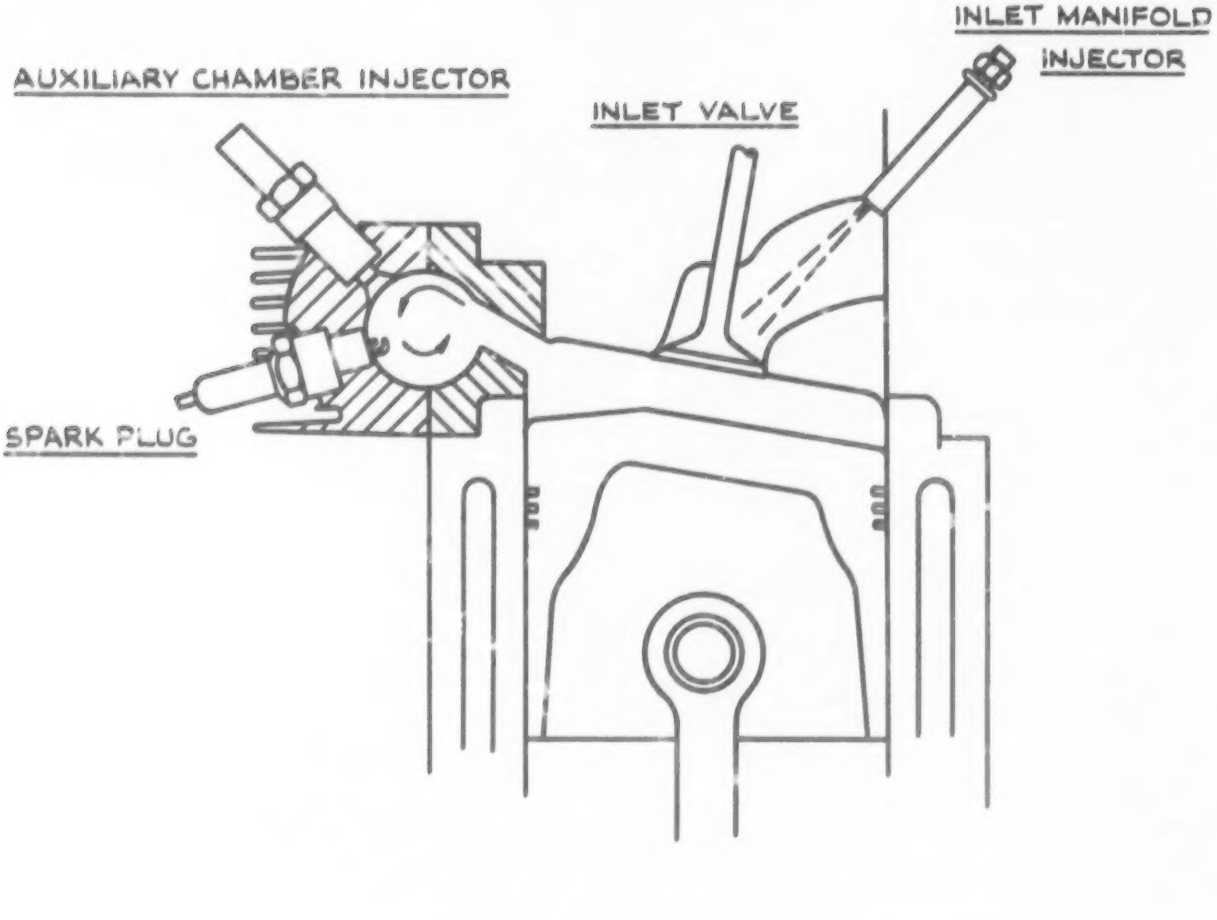
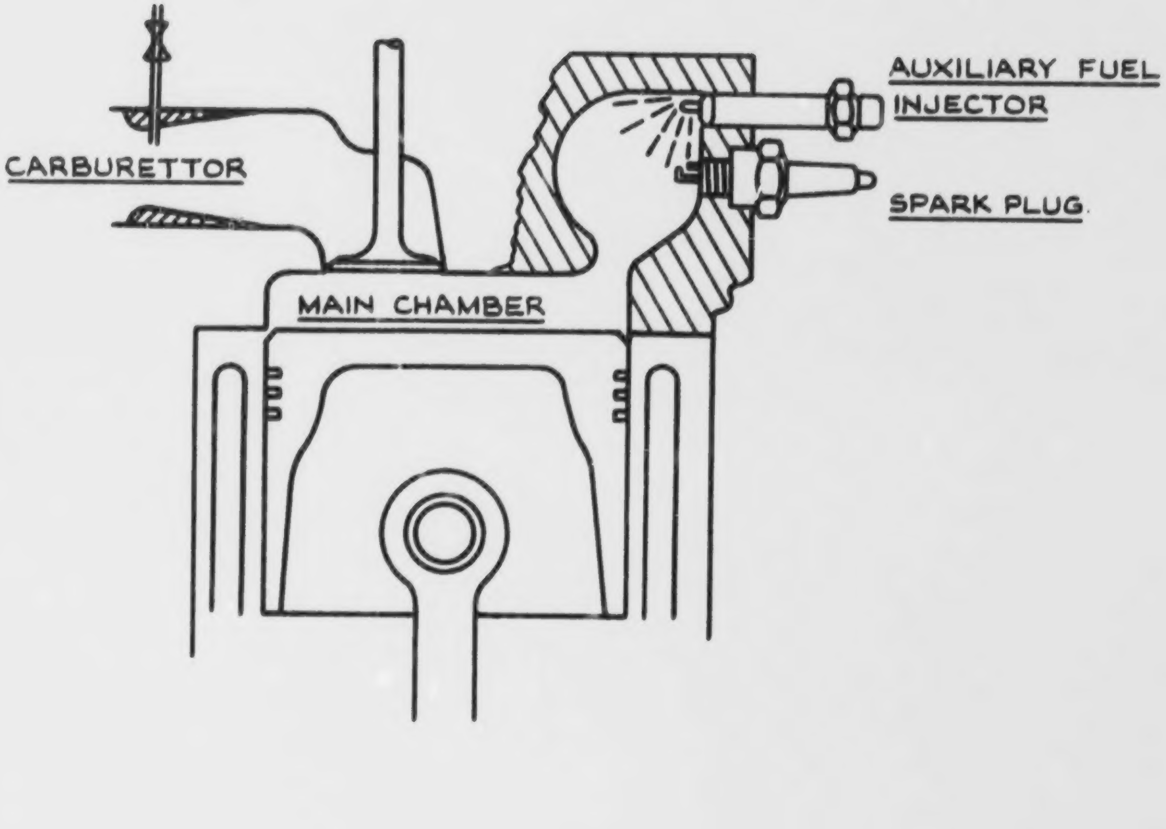

INLET MANIFOLD INJECTOR
AUXILIARY CHAMBER INJECTOR
INLET VALVE
SPARK PLUG

Figure 4.7. Schlamann.


AUXILIARY FUEL INJECTOR
CARBURETTOR
SPARK PLUG
MAIN CHAMBER

174

Figure 4.8. Pischinger.

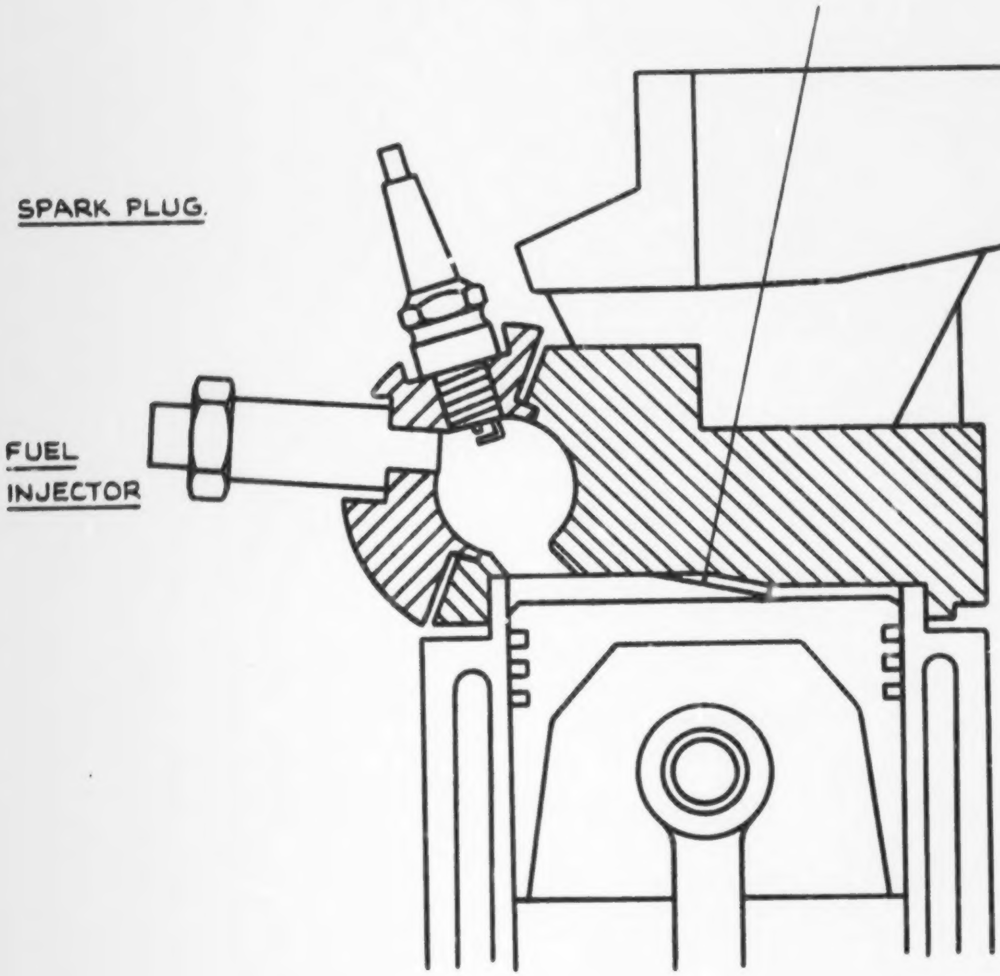

SPARK PLUG.
FUEL
INJECTOR

Figure 4.9. Huber.

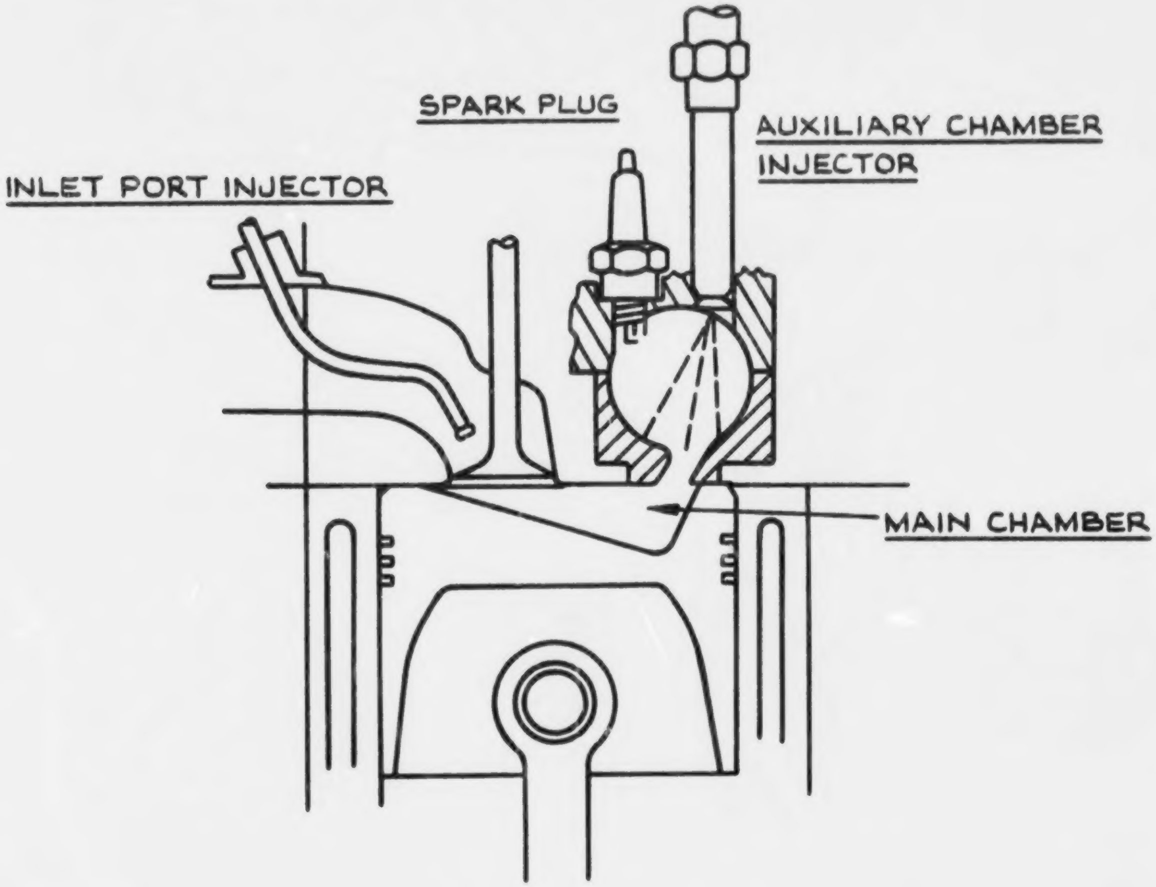

SPARK PLUG
AUXILIARY CHAMBER INJECTOR
INLET PORT INJECTOR
MAIN CHAMBER

Figure 4.10. Newhall.

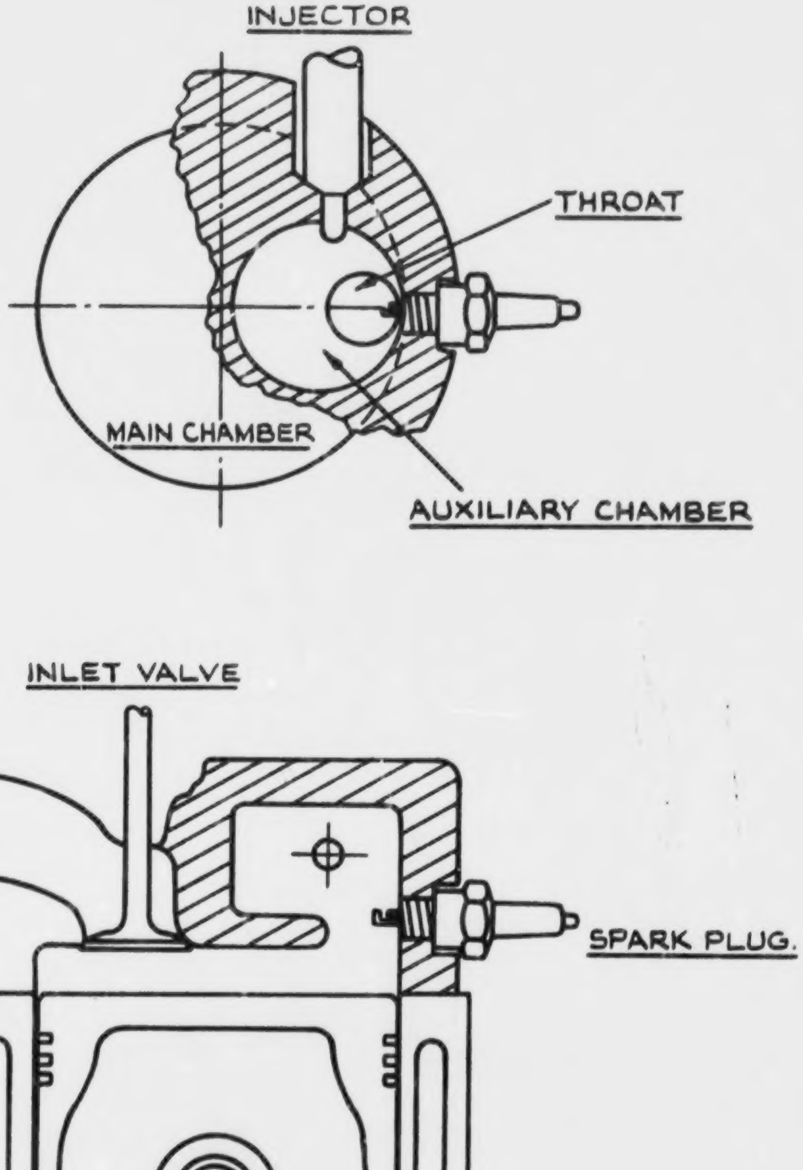

INJECTOR
THROAT
MAIN CHAMBER
AUXILIARY CHAMBER
INLET VALVE
SPARK PLUG.

Figure 4.11. Schlamann rubber pump.

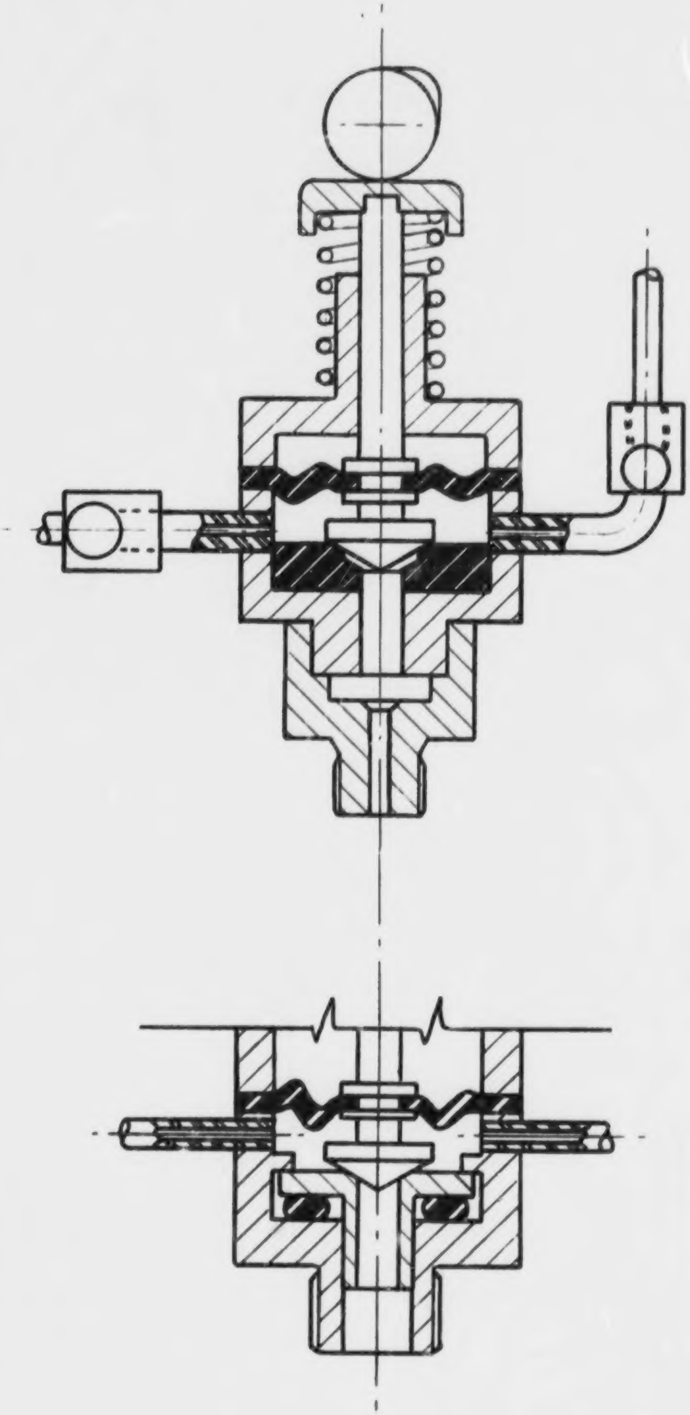


Figure 4.12. Maximum throat velocity of various prechamber engines during compression stroke.

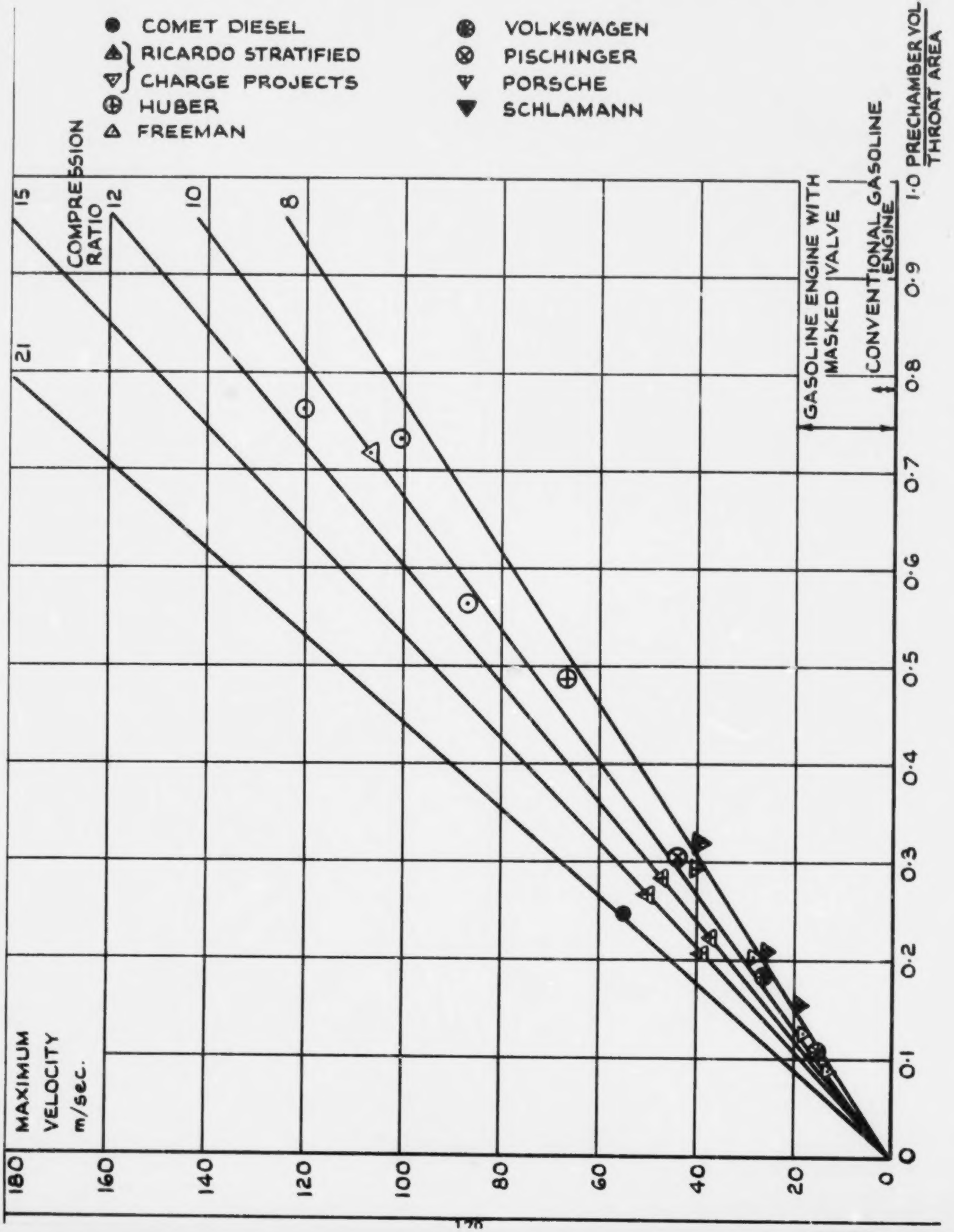


Figure 4.13. Throat velocity factor of prechamber engines.

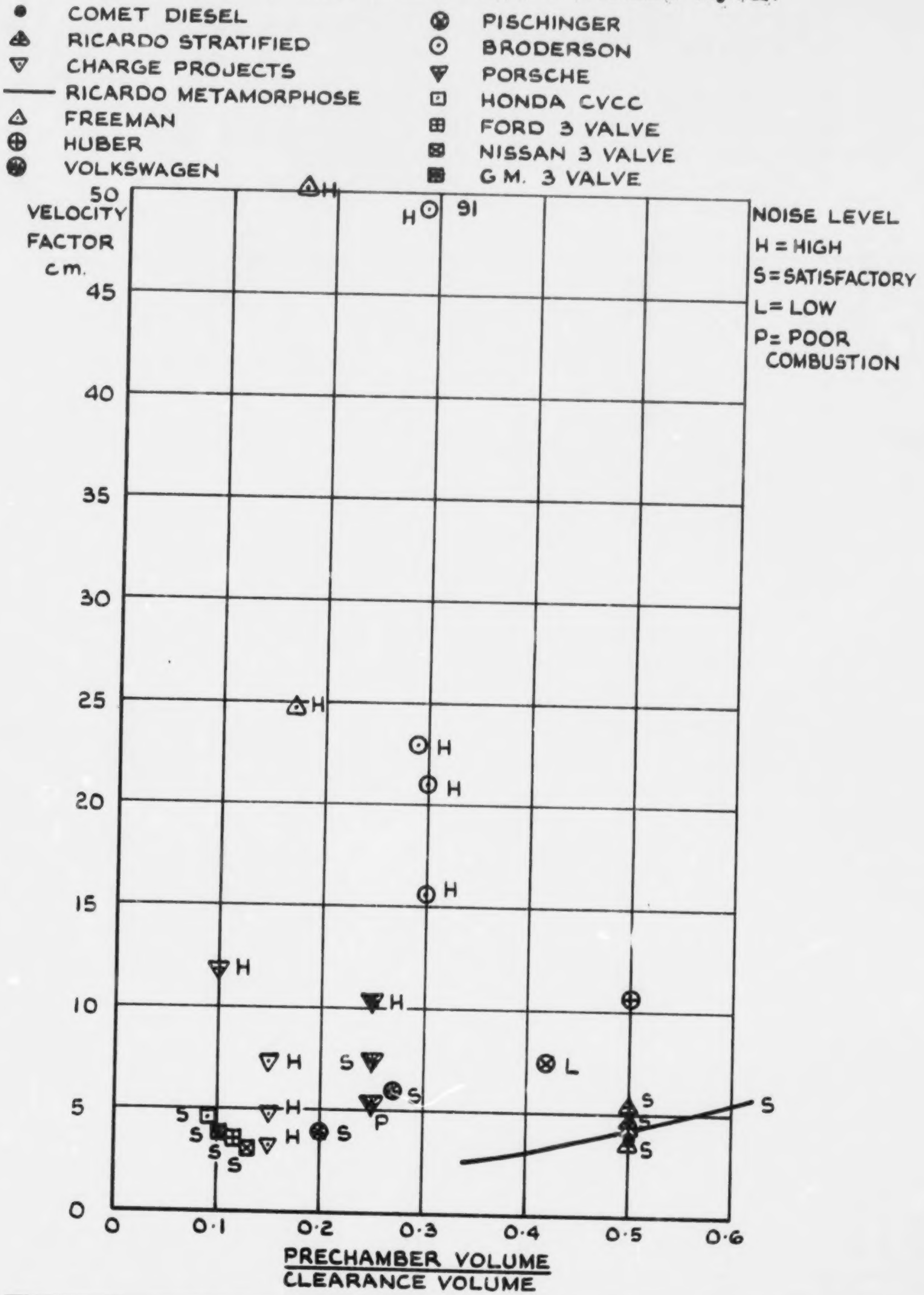


Figure 4.14. Cylinder pressure diagrams for some prechamber stratified charge engines.

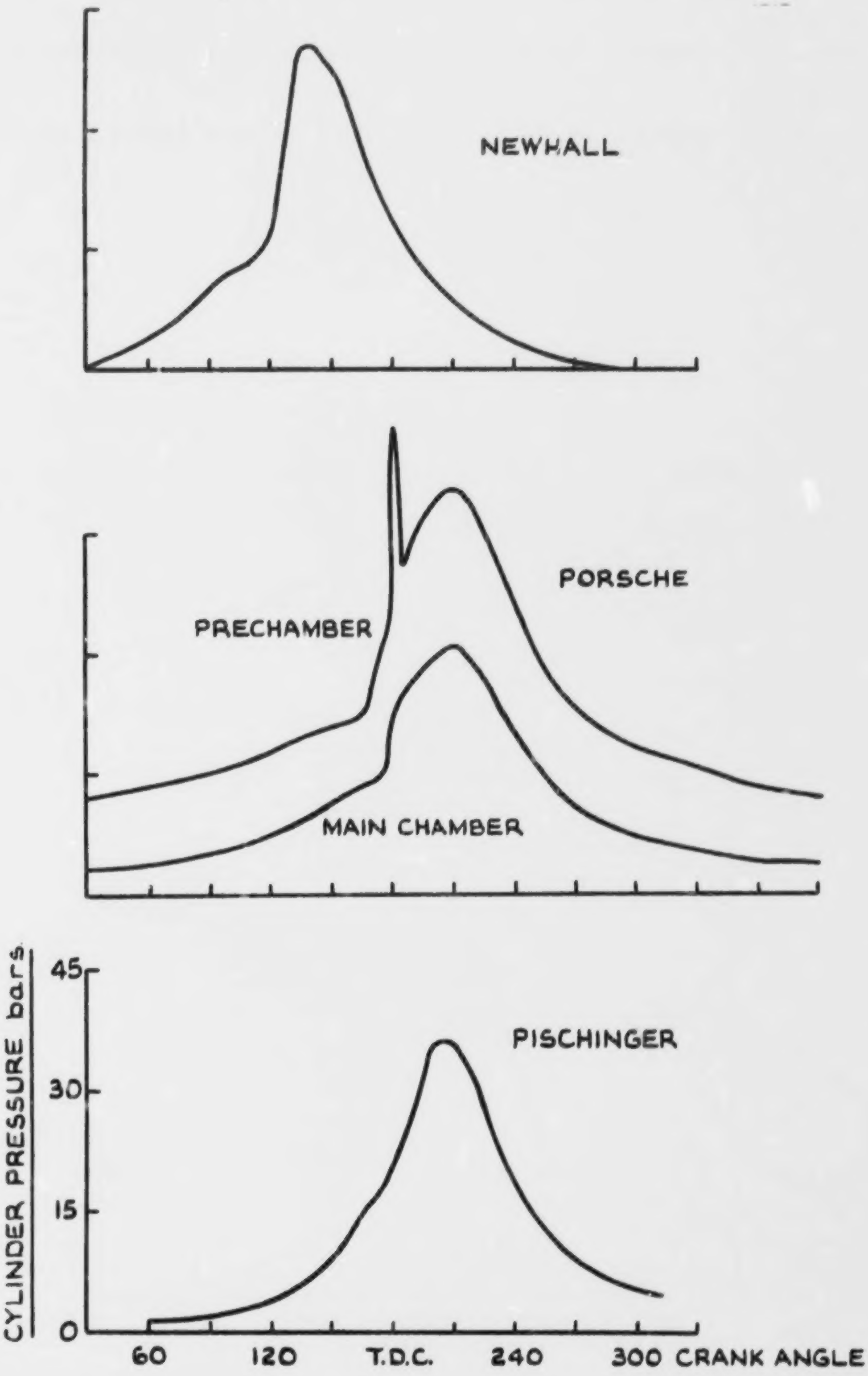


Figure 4.15 Comparison of torque curves VW stratified charge engine and baseline diesel (VW extrapolated from 4 cylinder 1.6 litre engine.)

V.W. S.C. ENGINE.

SWEPT VOL. FOR 96kW = 3·97 l ($242 in^3$) MAX. POWER SPEED = 4000 rev/min.

DIESEL ENGINE.

SWEPT VOL. FOR 96 kW = 4·77 l ($291 in^3$) MAX. POWER SPEED = 4000 rev/min.

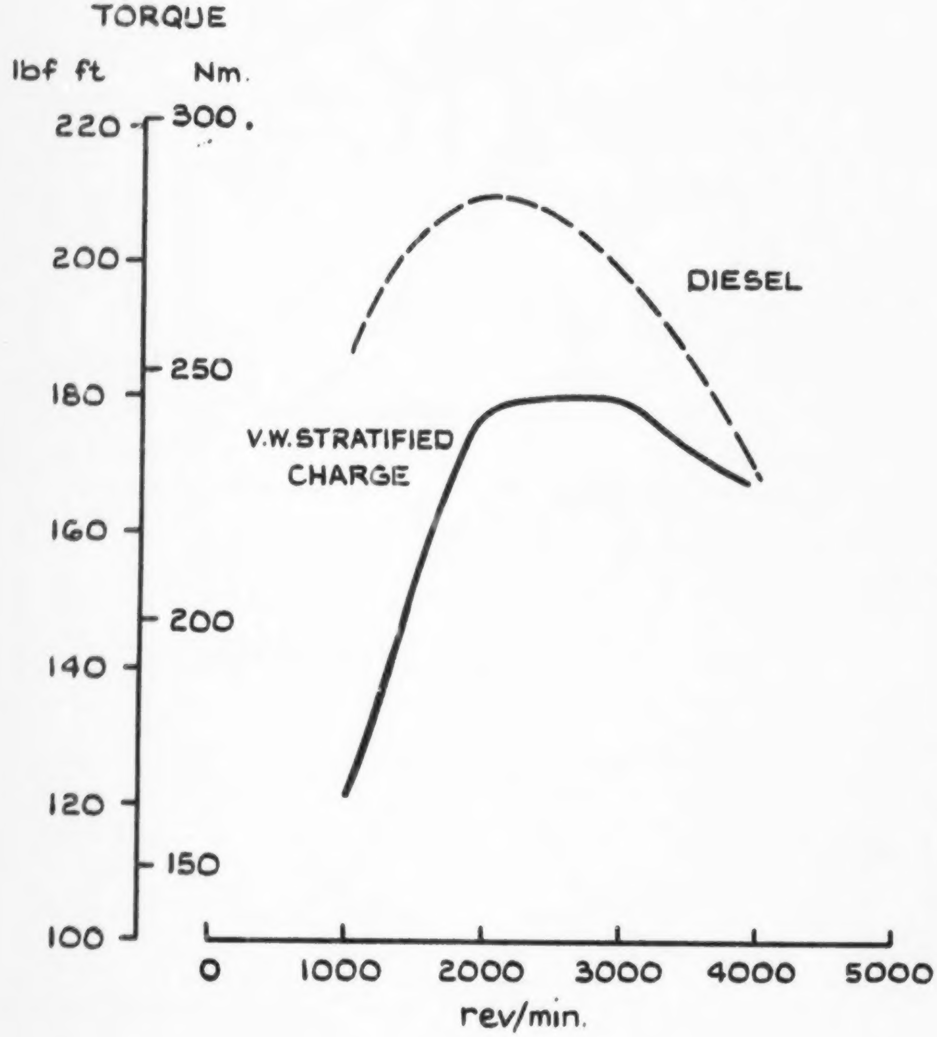


Figure 4.16 Indicated thermal efficiency vs fuel air equivalence ratio.

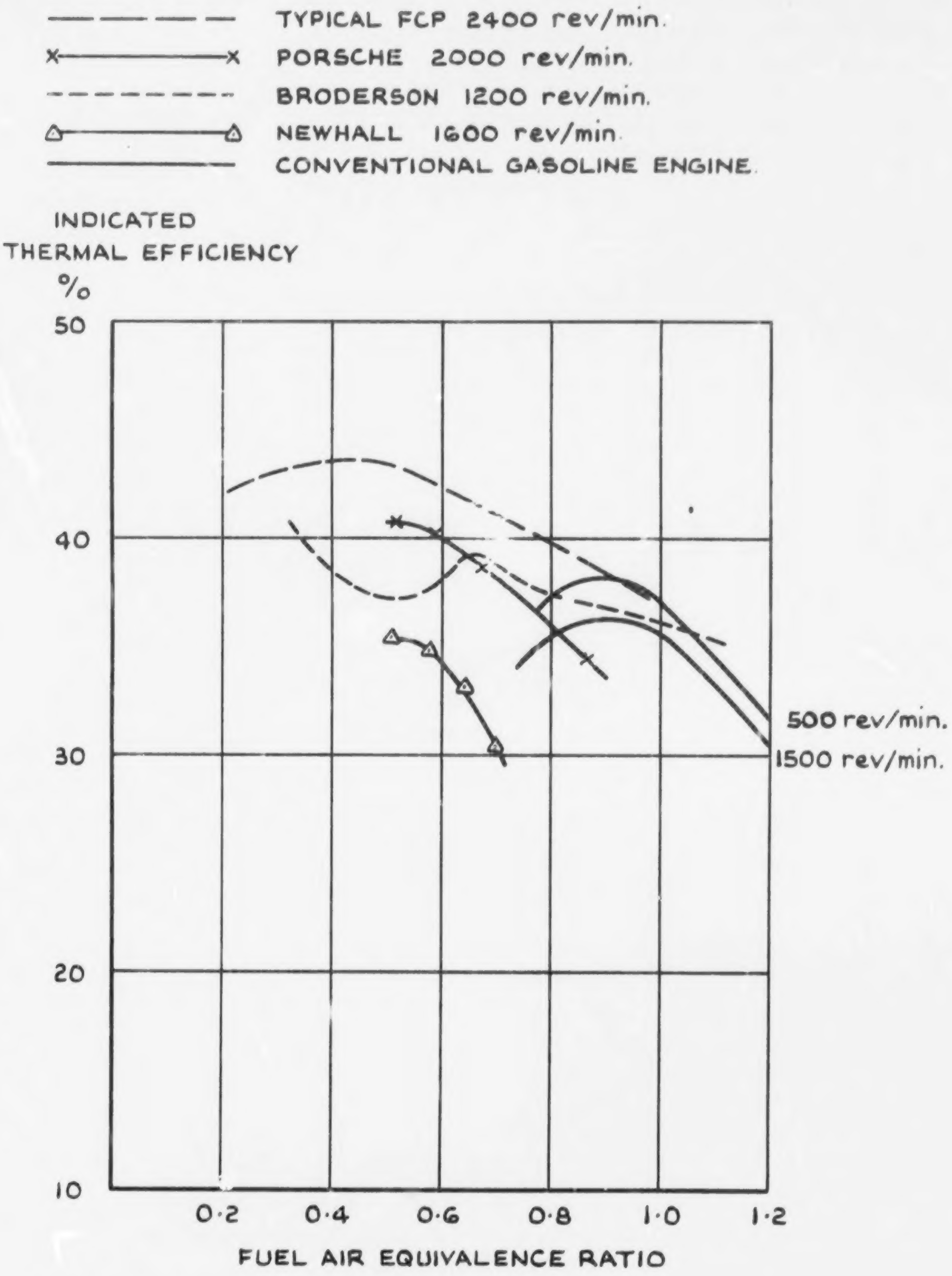


Figure 4.17. Indicated thermal efficiency vs I.M.E.P. for various stratified charge engines.

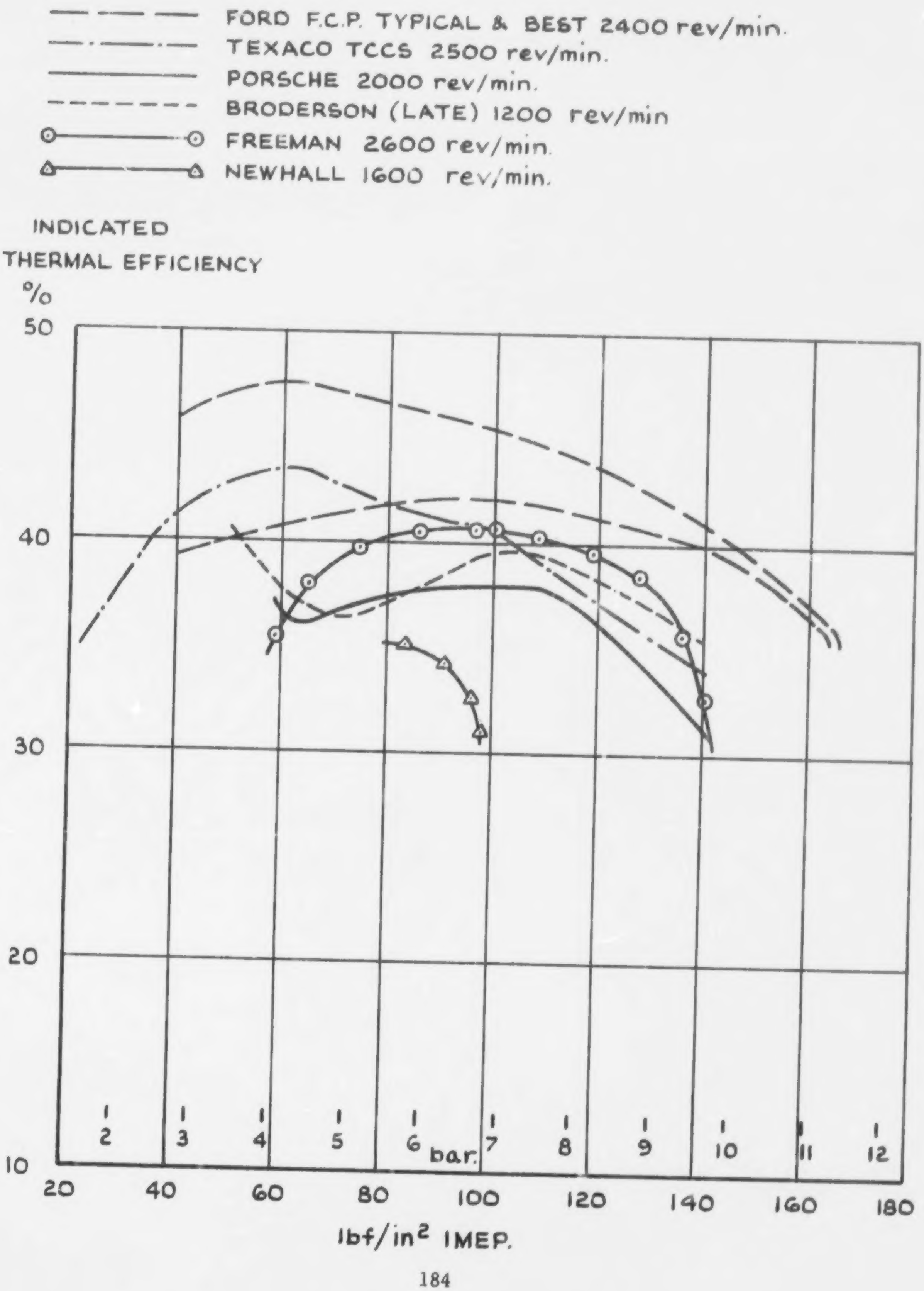


Figure 4.18 Comparison of emissions and fuel consumption of a Porsche prechamber stratified charge engine and a Comet Mk.V diesel engine.

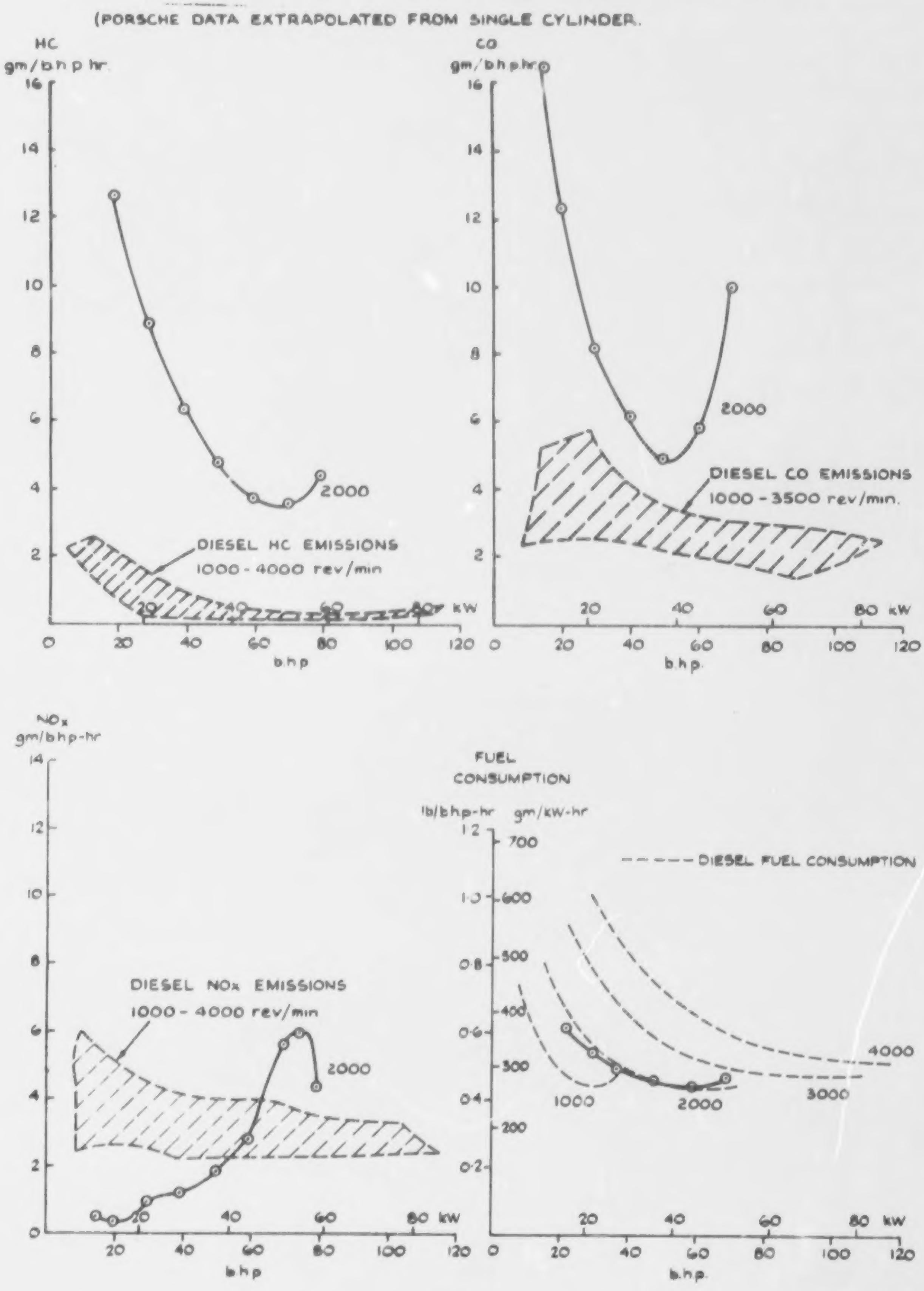


Figure 4.19. Emissions and fuel consumption of Pischinger stratified charge prechamber engine compared with a Comet Mk V diesel engine (stratified charge extrapolated from single cylinder results).

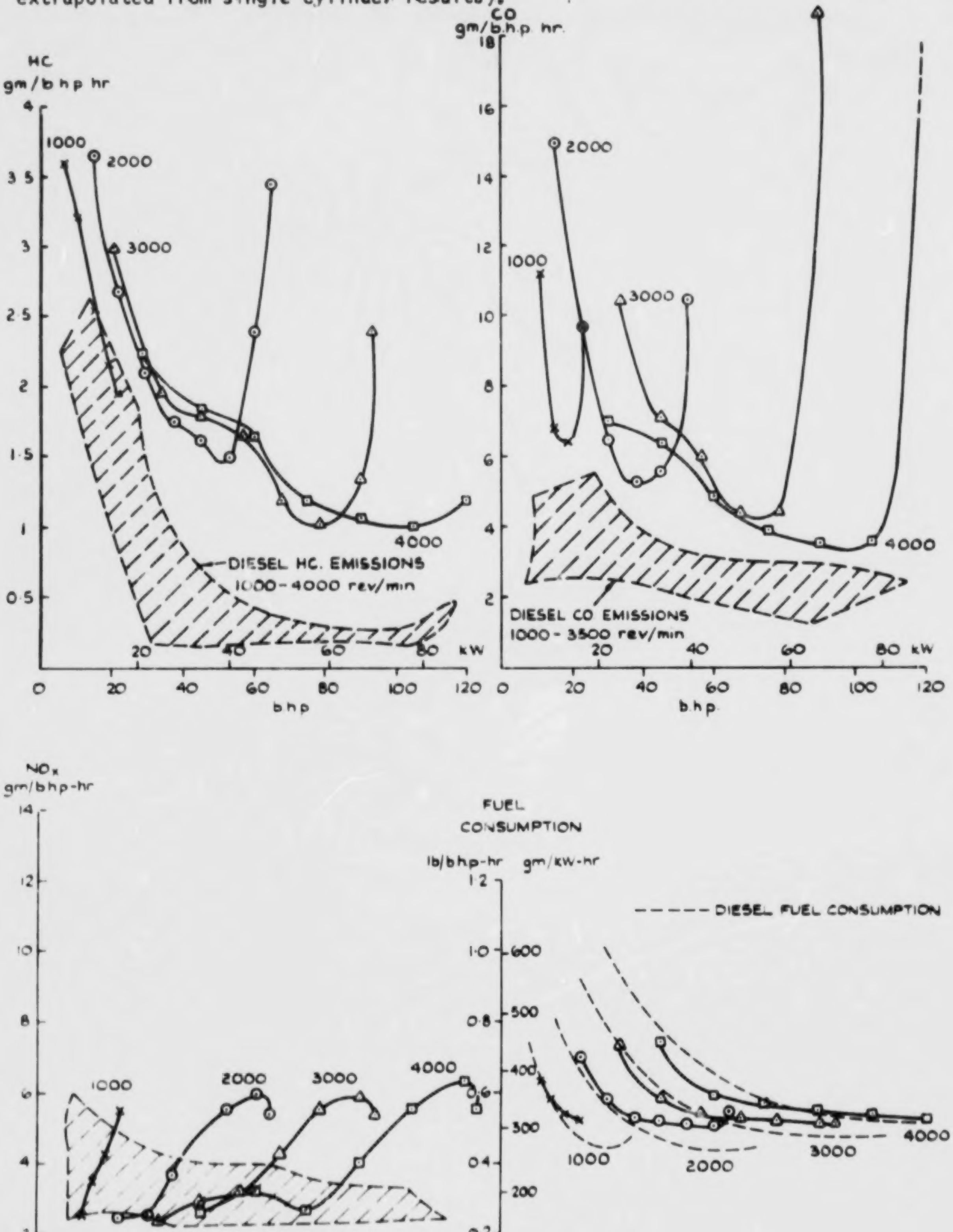


Figure 4.20 Comparison of emissions and fuel consumption of Volkswagen PCI stratified charge engine and a comet Mk.V diesel engine (VW results extrapolated from 1.6 litre 4 cylinder engine.

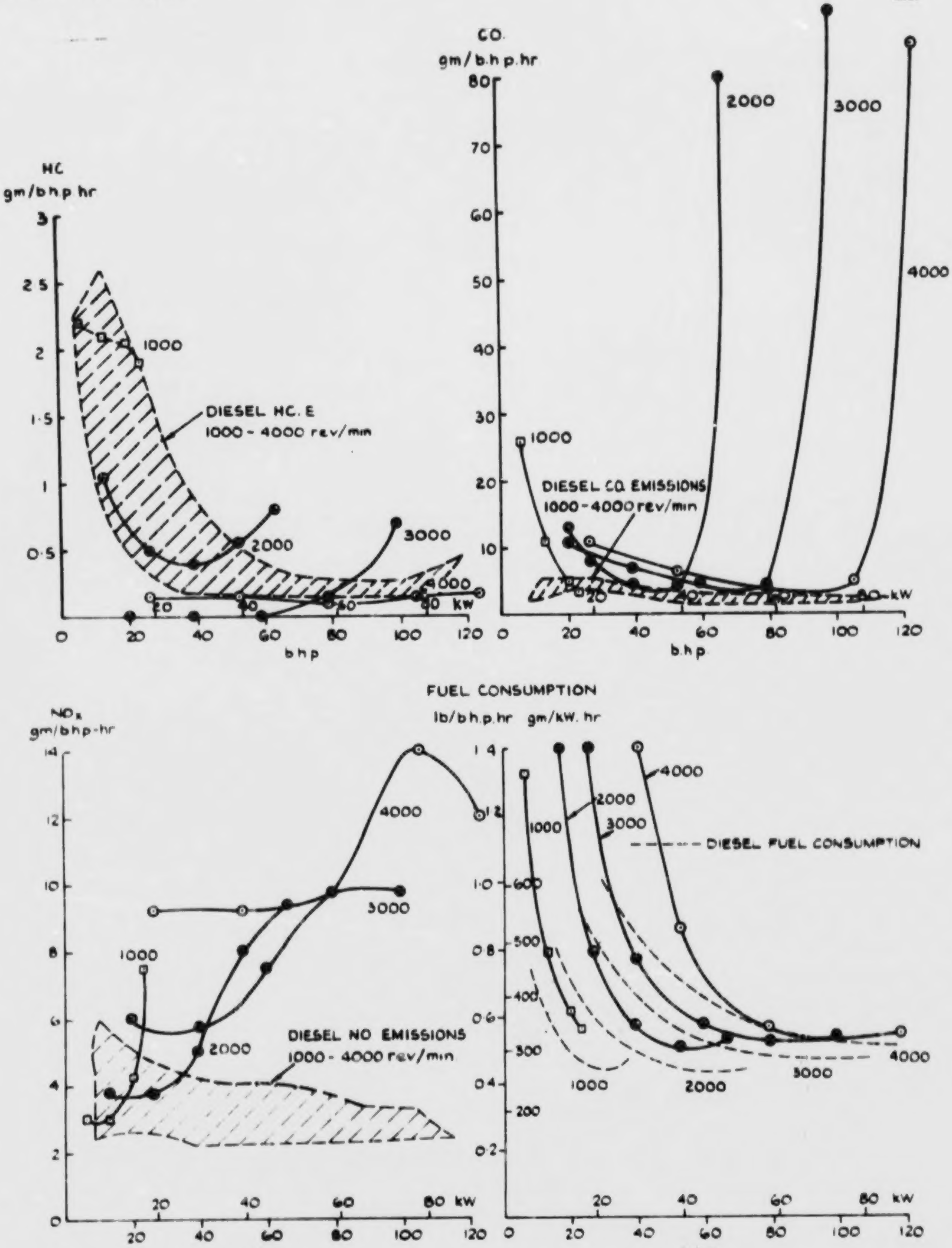
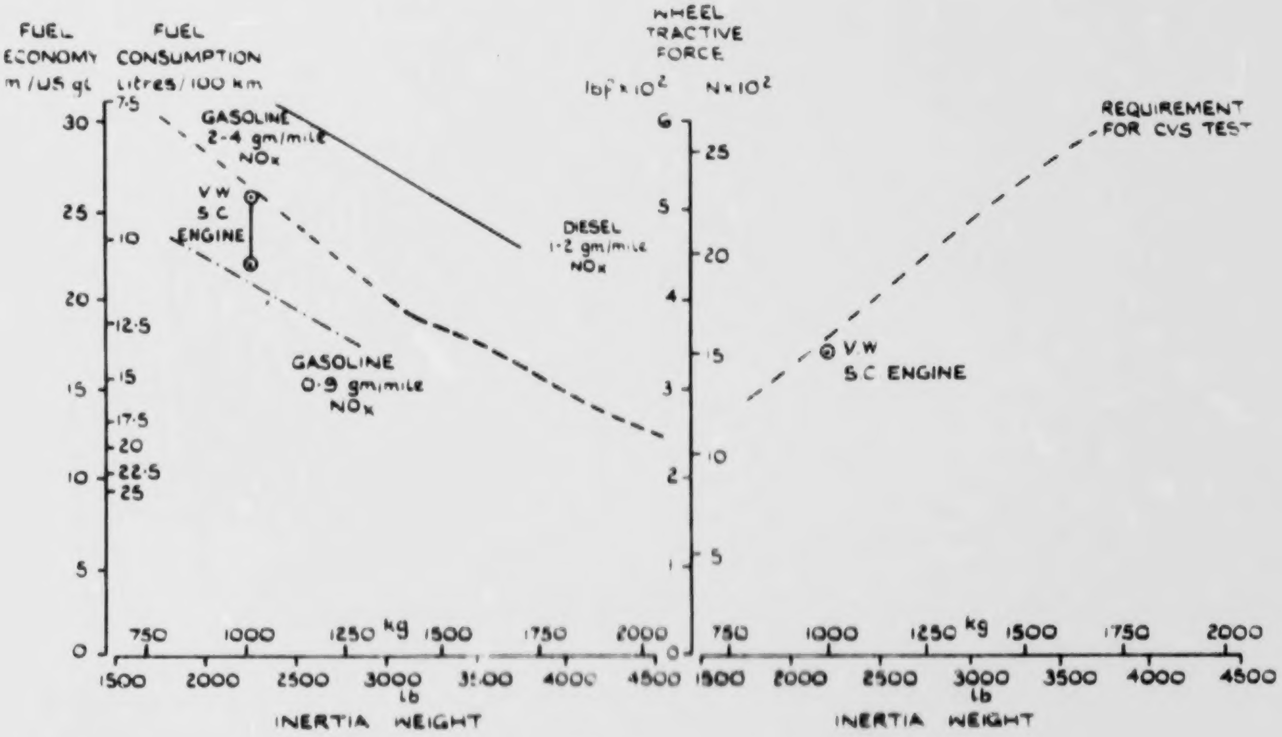
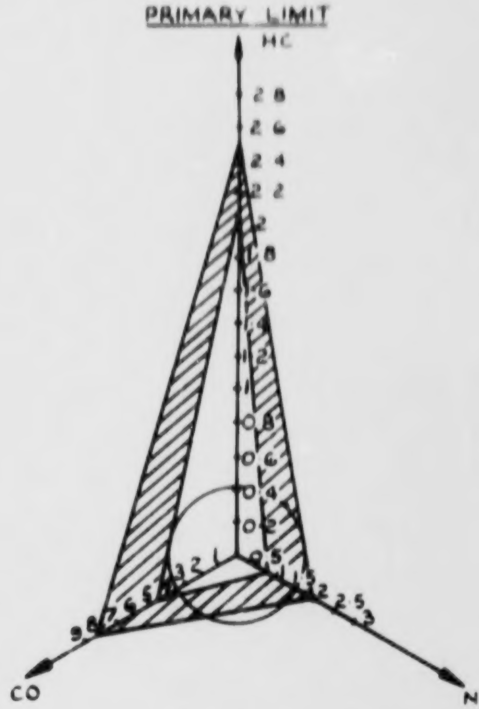


Figure 4.21. Emission results for a V.W. stratified charge engine without emission control devices.



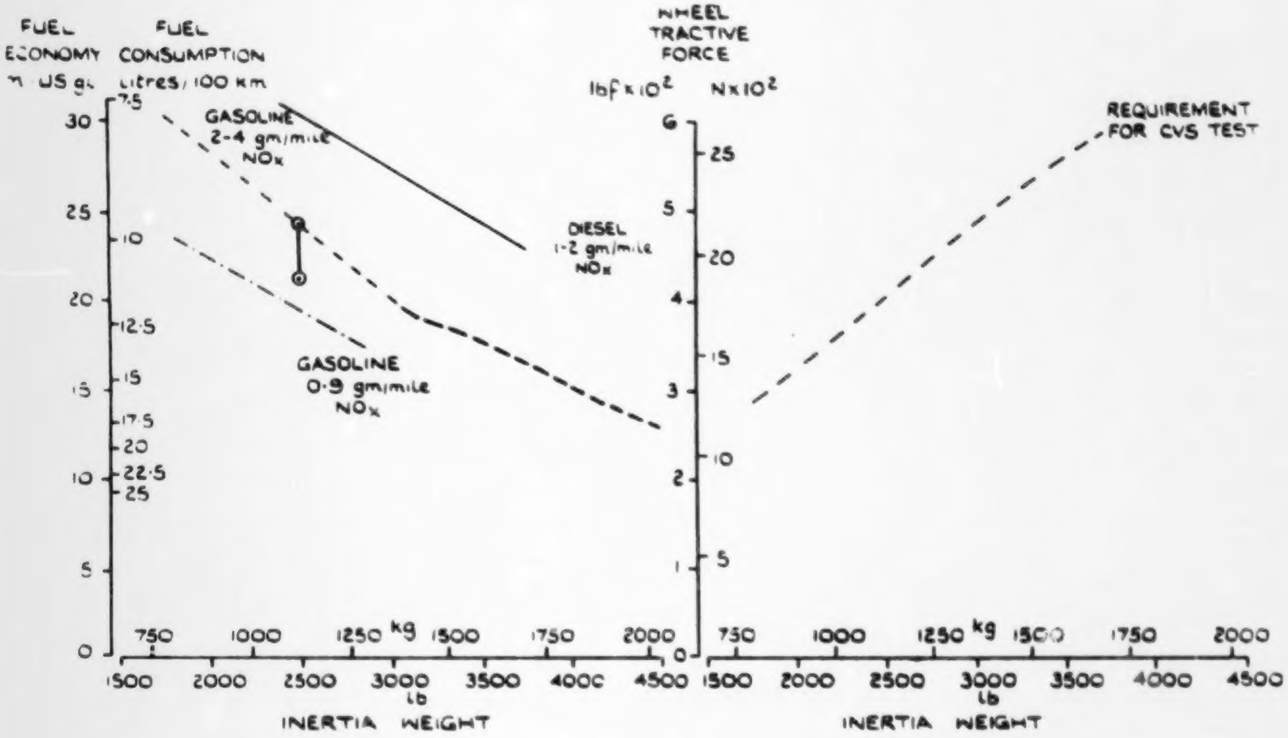


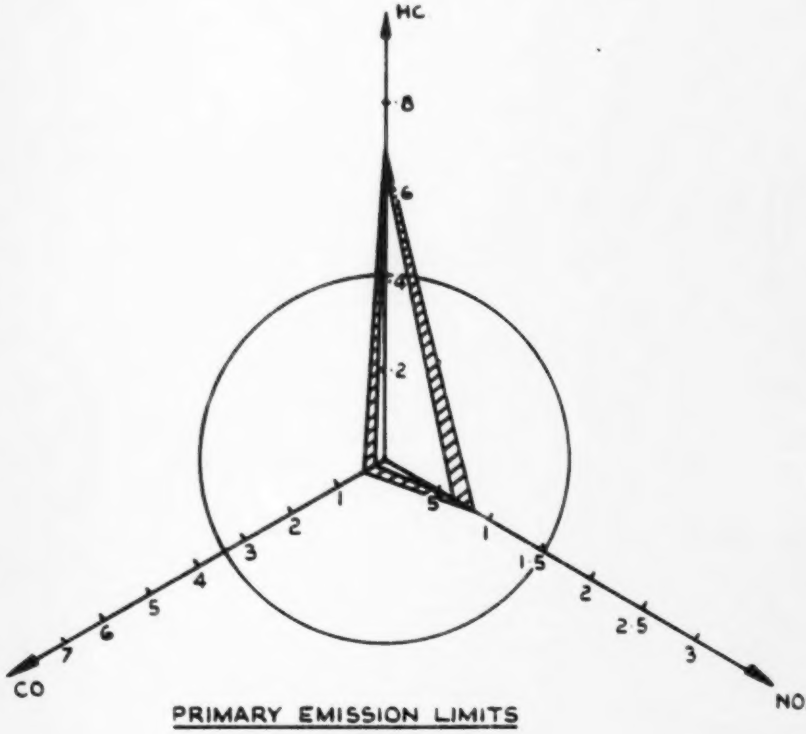
V.W. STRATIFIED CHARGE ENGINE

TESTED BY V.W.

TRANSMISSION MANUAL

Figure 4.22. Clawson concept stratified charge in Ford Capri fitted with catalyst.





PRIMARY EMISSION LIMITS

TESTED BY EPA

TRANSMISSION : AUTO

Figure 4.23. Emission results from a Clawson concept stratified charge engine. (Results extrapolated from 2.3 litre 4 cylinder engine.)

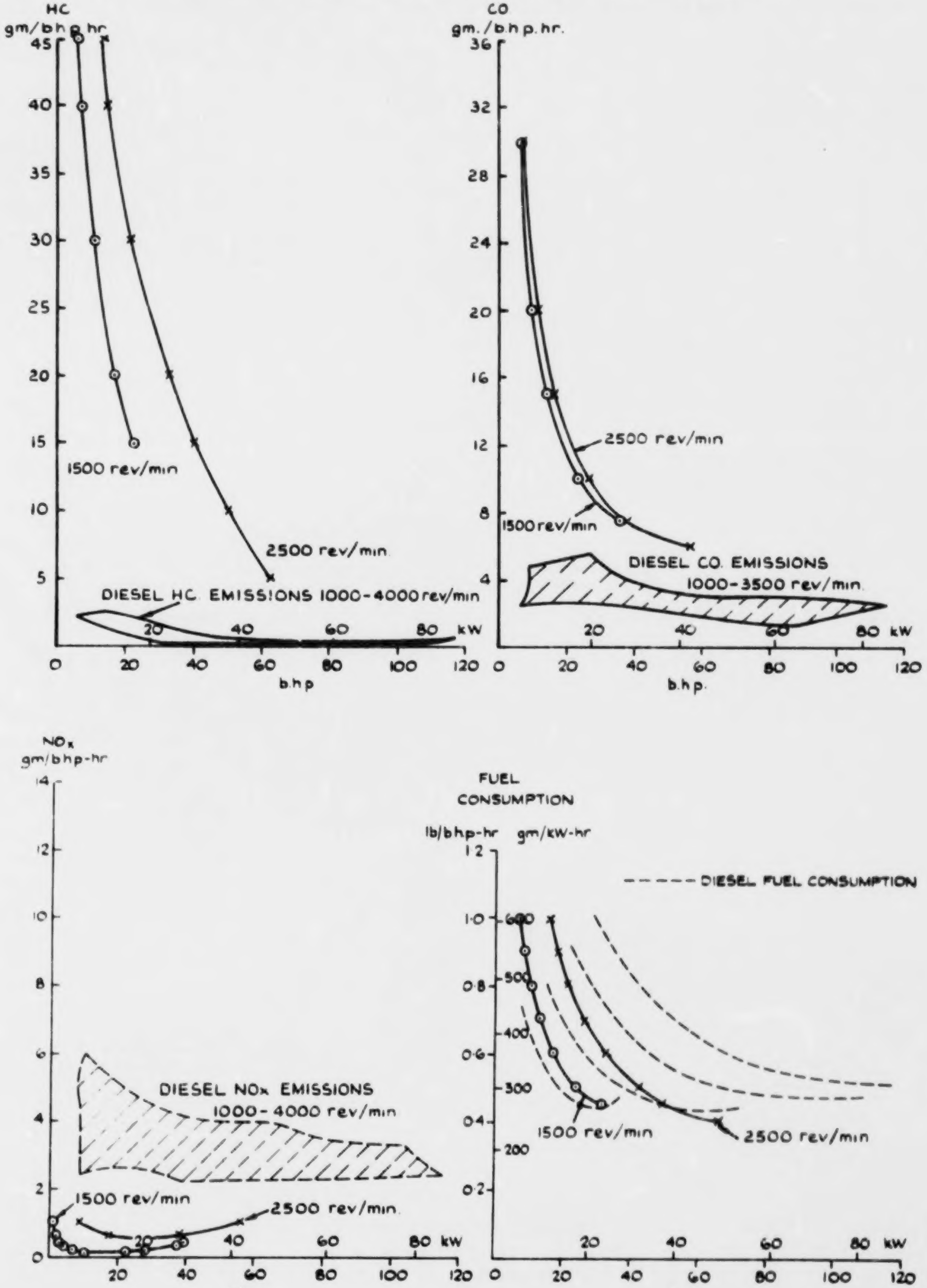


Figure 4.24. Emissions and fuel consumption of Huber multi-cylinder engine compared with a Comet Mk. V diesel engine. (Stratified charge extrapolated from single cylinder results.)

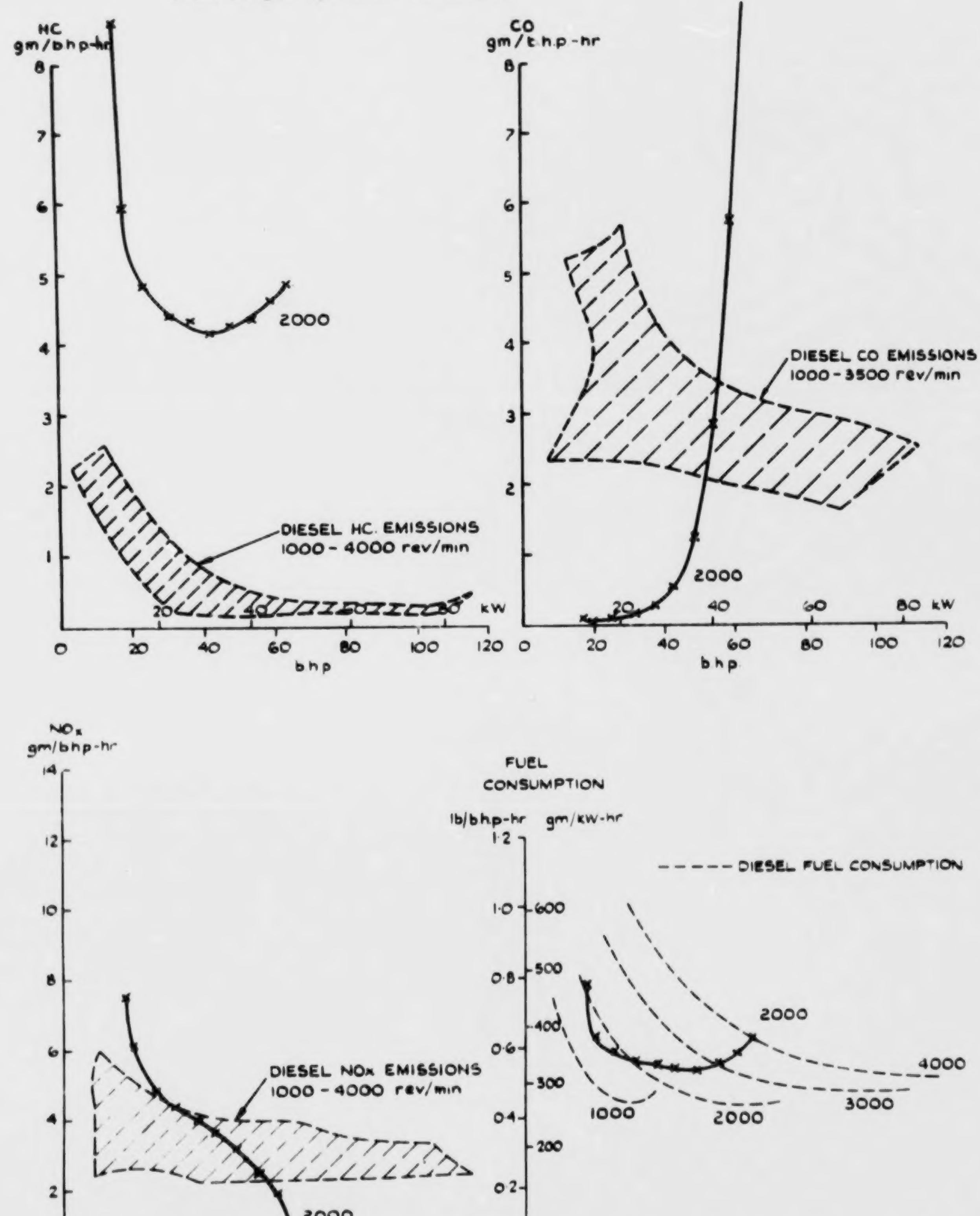
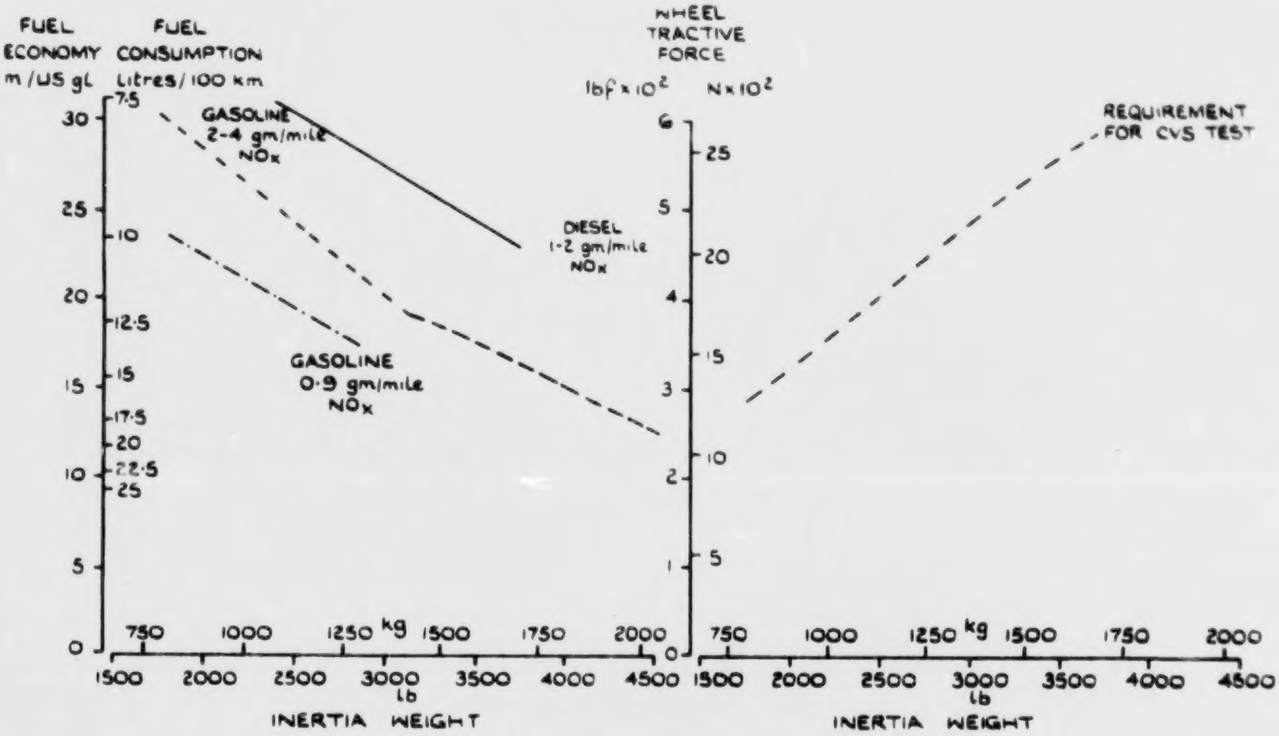
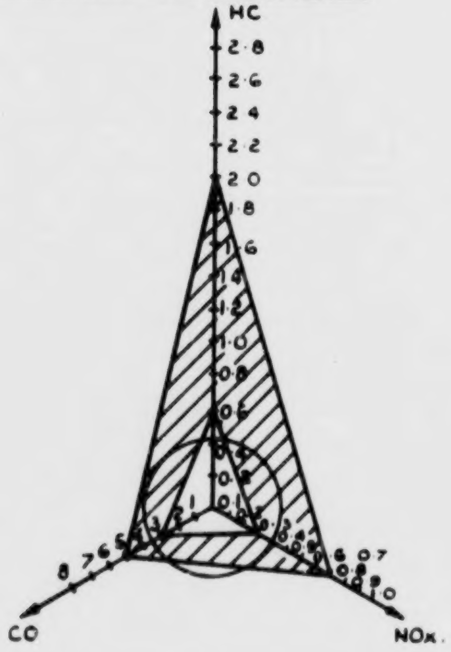


Figure 4.25. Emission results from a Porsche stratified charge engine. No fuel economy or torque data.



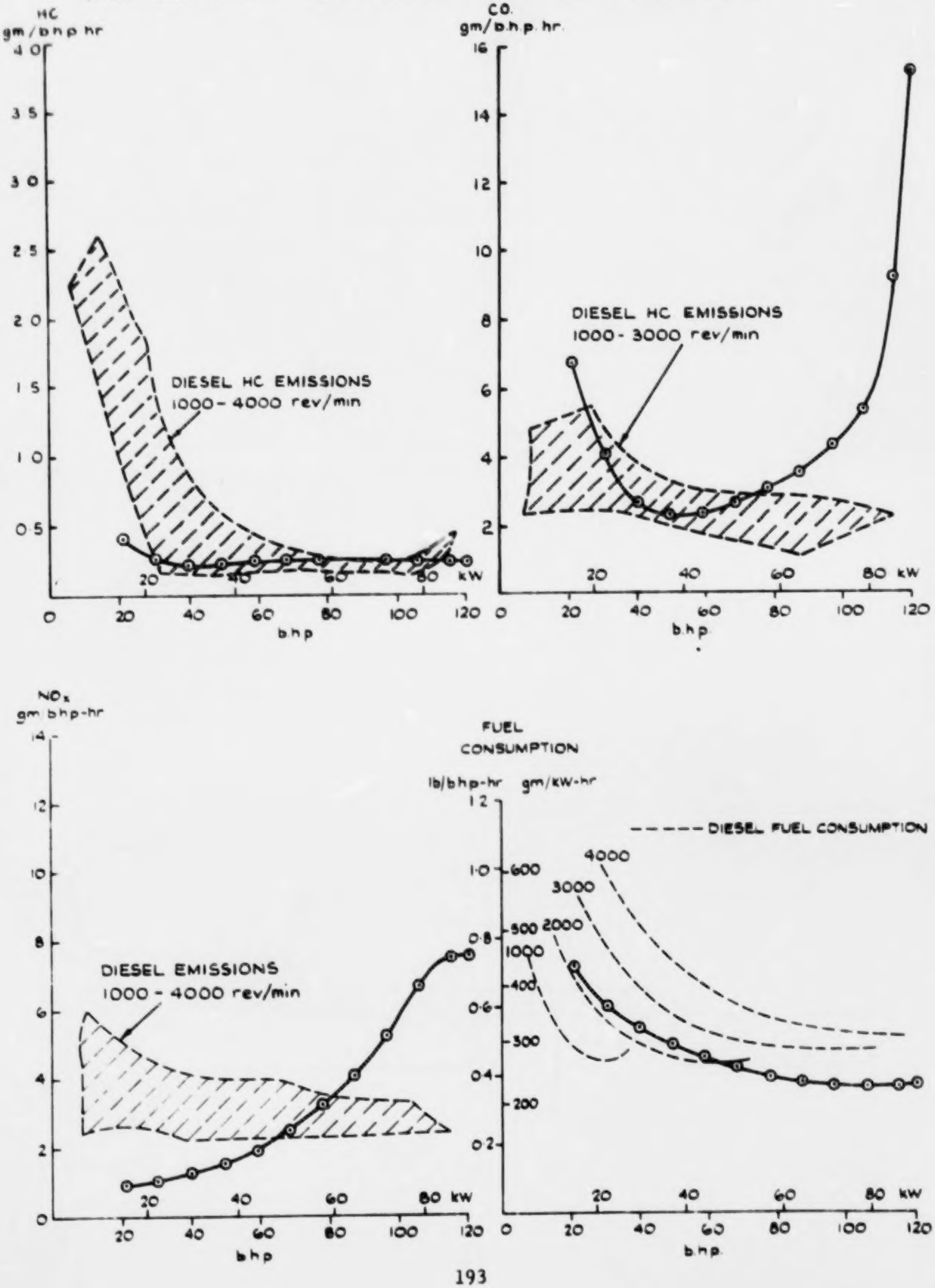
SECONDARY EMISSION TARGET



PORSCHE STRATIFIED CHARGE ENGINE

TESTED BY PORSCHE.

Figure 4.26. Emissions and fuel consumption of a Porsche stratified charge prechamber engine compared with a Comet Mk. V diesel engine.



ALL STRATIFIED CHARGE RESULTS AT 3000 rev/min

(EXTRAPOLATED FROM INDICATED SPECIFIC CURVES FOR 6 CYLINDER ENGINE)

Figure 4.27. Prechamber engine with pumpless injection.

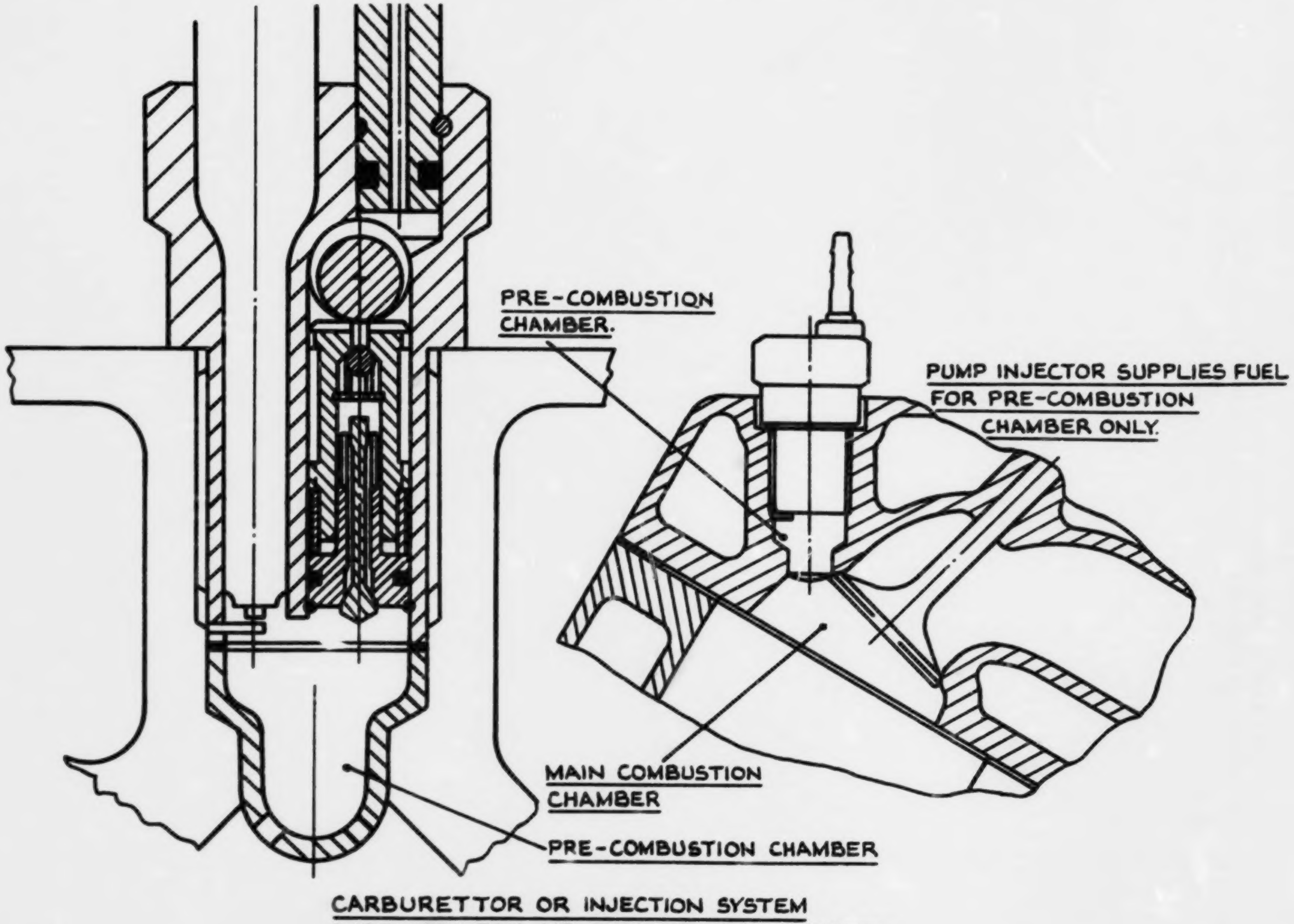
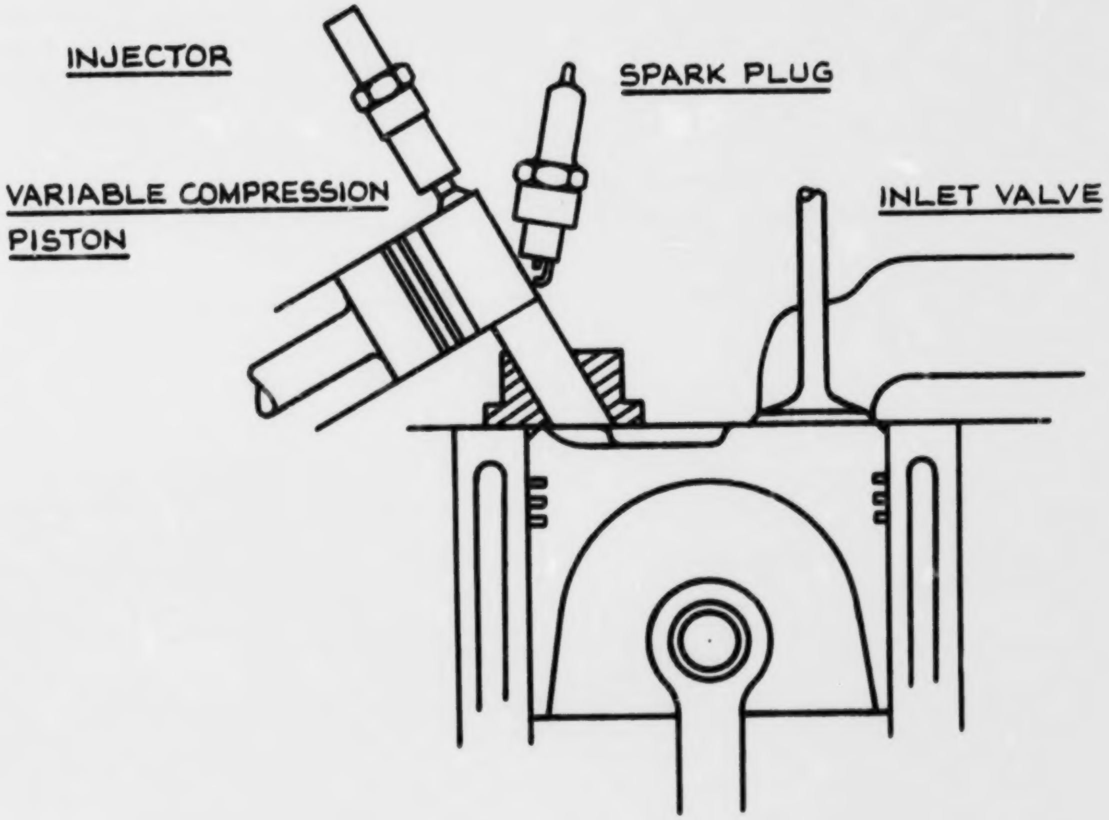

PRE-COMBUSTION CHAMBER.
PUMP INJECTOR SUPPLIES FUEL FOR PRE-COMBUSTION CHAMBER ONLY.
MAIN COMBUSTION CHAMBER
PRE-COMBUSTION CHAMBER
CARBURETTOR OR INJECTION SYSTEM SUPPLIES LEAN AIR-FUEL TO MAIN CHAMBER.

Figure 4.28. Ricardo "Metamorphose."


INJECTOR
SPARK PLUG
VARIABLE COMPRESSION PISTON
INLET VALVE

CATEGORY V

DIVIDED COMBUSTION CHAMBER WITH SEPARATE INLET VALVE

History

The scavenged prechamber engine where a rich charge, separate from the lean main charge is induced into the prechamber, is the oldest arrangement of stratified charge, dating from the 1920's. The basic idea has reappeared in a number of forms over the years, but the small efficiency advantages were never significant enough to warrant the extra engine production costs. In particular, many Russian experimenters tried to optimise the economy on low compression ratio engines. Recently, this situation has altered dramatically, as the Honda Motor Company demonstrated that the scavenged prechamber engine could be tailored to give lower exhaust emissions than current gasoline engines.

Topic Area (a) Combustion Chamber and Stratification Technique

The various studies on scavenged prechamber engines can be conveniently divided into four groups, depending on the time at which the experiments were carried out, and the goals of the study. These are:

1. Systems developed recently, following the same approach as the Honda CVCC System.

2. Work carried out in Russia in the 1960's and 70's, with the aim of improving fuel consumption.

3. Various retrofit systems for conventional gasoline engine, using prechambers with automatic valves, which screwed into existing spark plug holes.

4. Other miscellaneous work carried out in the past which does not fall into the above groups.

1. Recently Developed Scavenged Prechamber Engines Ref. 5.1.

A number of companies have published results which fall into this group but the combustion chamber designs and methods of operation are all very similar. The Honda CVCC fig. 5.1 is a typical design. The engine induces a lean mixture through the main intake valve, and a very rich mixture through the prechamber valve. During the compression stroke, the mixture in the prechamber is diluted by some of the main chamber mixture, so that an ignitable mixture of an A/F ratio between 9 and 12 occurs at the spark plug at the time of ignition. After ignition by the plug, the hot burnt gas in the prechamber is expelled as a torch across the main combustion chamber, to initiate combustion in the lean mixture of the main chamber. The expansion stroke then proceeds as usual.

This description may be over-simplified, as some of the references maintain that the stratification of the mixture is relatively complex, and cannot be considered as simply a rich prechamber and a lean main chamber mixture. During the induction stroke, the amount of mixture drawn through the prechamber is often many times greater than the volume of the prechamber. Therefore, some rich mixture will overflow through the prechamber orifice into the main chamber. The rich mixture will partially mix with the main chamber mixture, forming a "mixture cloud" with an air/fuel ratio somewhere between that of the pre and main chambers. The extent of the mixing of pre and main chamber charges is a function of the volume flow through the orifice and the orifice size and shape.

The latter design factors determine the velocity and turbulence of the charge as it enters the main chamber. During the compression stroke, a proportion of the mixture cloud is forced back into the prechamber. According to Honda, the remainder of the mixture cloud which remains in the area of the throat, plays an important part in adding fuel to the igniting jet as it emerges from the prechamber. Therefore, many factors influence the distribution of the fuel within the two combustion chambers.

The amount of mixture entering the prechamber is obviously important, and is usually expressed as a ratio of the prechamber volume, called "number of prechamber fills". The area of the orifice is also usually expressed in a ratio to the prechamber volume. A list of the important design and operating parameters used in the references is included in Table 5.1.

The second Honda engine refers to the 1.5 litre Civic car which is now in production. It is interesting that the GM and Honda publications disagree as to the importance of the "mixture cloud", and each have verified their views with in-cylinder sampling. In engine tests, both GM and Honda varied the number of prechamber fills, and while Honda observed a distinct minimum in HC, CO and NO_x emissions and fuel consumption, GM observed little effect except for a slight peak in NO_x. The reason for this discrepancy is not obvious, although the cavity between the orifice and the main combustion chamber of the Honda and Ford engines may help to retain the mixture cloud, Fig . 5.1. The absence of a prechamber overflow on the GM and Nissan engines will lead to different stratification within the cylinder. (Fig. 5.2).

GM, Ford and Honda all use open main combustion chambers without squish, although Eaton have found that some intake swirl helps to retain the integrity of the mixture cloud, Fig. 5.3. In many of the engines, the precombustion chamber is formed within a separate thimble, which contains the main orifice. Thus, there is an air gap between the prechamber and the cooled metal components. The insulation of the prechamber is arranged for two reasons:

1. heat transfer is reduced and efficiency improved.
2. liquid fuel entering the prechamber during a cold start is quickly vaporised. If it did not vaporise, the stratification would be affected. Fuel vaporisation in scavenged prechamber engines is of critical importance.

If the rich prechamber mixture were not heated, then only a small proportion of the fuel would evaporate and mix with air. It would be very difficult to divide the liquid fuel in the manifold of a multi-cylinder engine, such that each cylinder received an equal share. The obvious solution is to heat the prechamber mixture until all the fuel is evaporated, and this approach is adopted universally in these engines. The main chamber mixture is also usually heated. The prechamber manifold temperature used by VW was 100°C, Fig. 5.2, and by Ford 200°C. The temperature of the main charge of the Ford engine was 82°C at low load conditions.

The references do not contain information on the effect of prechamber valve timing on the engine operation. Some have chosen to operate the prechamber valve from the cam of the main inlet valve so that the timings were identical while others have adopted separate cams. The Honda CVCC production engine used separate cams and valve gear for actuating the prechamber valve.

2. Russian Scavenged Prechamber Engines. Ref. 5.6.

There are few details on the design of the Russian engines, although the general concept is certainly similar to the Honda process. The main developments have been by Nilov, who used a very small prechamber, and the GAZ-52 and ZIL-120FK which were 4 cylinder production engines. The prechamber of the ZIL engine occupied 3-4% of the clearance volume. The main difference between the Russian engines and the current developments in Japan and the West are:

(a) The prechamber valve is always actuated by the main inlet rocker.

(b) The prechamber does not usually contain a thimble.

(c) The main combustion chamber usually incorporates a squish area for charge movement; 23% of the piston area in the case of the ZIL engine.

3. Retrofit Devices. Ref. 5.5

For some years, the development of emission control technology for new engines has been paralleled by efforts to control the emissions of existing vehicles on the road, by the use of retrofit devices. One popular approach has been to install a prechamber on an existing engine. The prechamber is screwed into the spark plug hole, and rich mixture from a carburettor is induced through an automatic valve. A spark plug is installed in the prechamber. The main chamber carburettor can then be set leaner as the prechamber mixture will initiate combustion in a jet ignition manner as in the Honda engine. The prechamber usually occupies around 3% of the combustion chamber volume. This general approach has been followed by Morghen, Phillips, Teledyne and Walker. All their efforts have been unsuccessful due to the limitation of maximum speed of operation of the automatic valve and poor HC emissions.

An example of this prechamber design, i.e. the Walker Stratofire is shown in Fig. 5.4.

4. Early Scavenged Prechamber Engines Ref. 5.9, 5.1.7 and 5.5.2.

Research on scavenged prechamber engines has been carried out at the Ricardo laboratories on a number of occasions since 1920. One interesting variation on the recent Honda system was to close the main inlet valve relatively early, and open the prechamber valve immediately afterwards. The prechamber tract was supercharged or turbocharged, so that overall volumetric efficiencies greater than 100% could be obtained. The great advantage of this arrangement is that the total induced charge does not need to pass through the compressor, but only that needed for topping up the cylinder.

A study of jet ignition for improving fuel economy was made in the 1960's by Rodgers, ref. 5.1.7. but the gains identified were minute. The prechamber was not fed by a separate carburettor but was arranged to collect liquid fuel entering the engine cylinder. The engine was designed to give very high ignition jet velocities, in the region of 200 m/sec.

The design of the combustion chamber of the Heintz engine is different to that of other scavenged prechamber engines, Fig. 5.5. The prechamber is arranged in the centre of the main combustion chamber, with six orifices directed radially into the main combustion chamber. In addition, a collector is arranged in the prechamber to prevent liquid fuel entering the main combustion chamber. Fuelling is provided by port injection for both the prechamber and main chamber inlet tracts. This engine is usually operated unthrottled, and the load is controlled by the fuel flow to the main chamber port injector above 40% load. Below this load, air is induced into the main chamber, and the prechamber fuelling is used to control the load.

Carburettors.

The charge supply to the engine is usually prepared by carburettors, although some investigators have used fuel injection for experimental purposes. The manufacturing tolerances on carburettors for scavenged prechamber engines are more critical than on conventional engines due to the leaner mixture strengths. There are two reasons for this:

(a) If a tolerance of ± 1 air/fuel ratio is permitted at an air/fuel ratio of 20, then the maximum variation in fuel flow is 10%. However, the same air/fuel ratio tolerance at an air/fuel ratio of 15 allows 13.3% variation in fuel flow. Therefore the fuel flow tolerance of the lean burn engine is less than for an engine operating at stoichiometric conditions.

(b) The mixture strength corresponding to the best fuel economy and minimum HC emissions are virtually identical for the conventional gasoline engine, i.e. around 16 - 17 air/fuel ratio. If an engine is tuned for this condition, small variations in mixture strength will only cause minor deterioration in HC and fuel consumption, as the characteristic curves are relatively flat. However, the overall air fuel ratios corresponding to minimum consumption and minimum HC emissions are 21 and 17 respectively for the scavenged prechamber engine, Table 5.1. It follows that a compromise mixture strength must be chosen say 19:1. If the carburettor setting strays from this figure, then either HC emissions or fuel consumption will improve while the other will deteriorate significantly as it is situated on a rising characteristic curve. Therefore, the tolerance in air/fuel ratio for the prechamber engine is less than for the conventional gasoline engine.

The operation of the scavenged prechamber engine demands that a relatively large volume of charge is drawn through the prechamber inlet valve. Gas viscous flow effects are greater in the prechamber induction system than the main induction system. It follows that the throttle blade in the prechamber tract should be opened further than the main induction system. If this is verified, then it might be possible to control the throttle of the prechamber with a pressure capsule sensitive to the difference in manifold depression. At the moment, Honda have simply controlled the throttle with a mechanical linkage to give a constant angular difference between the throttle blades.

Ignition.

The positioning of the spark plug within the prechamber is important, so that the liquid fuel flowing in through the prechamber valve during cold starts does not impinge on the gap and prevent ignition. In addition, the plug gap should also avoid direct impingement of the gas inflow during the compression stroke, or else the mixture at the plug will be leaner, and the turbulence greater than in the remainder of the chamber.

The normal coil ignition system is apparently satisfactory to initiate combustion in these engines and high energy systems are not required.

The distributor of the Honda engine incorporated the normal speed and load modulation facilities for ignition timing, as well as an additional facility to advance the timing during cold starts, depending on the coolant temperature.

Octane Requirements.

The Russian investigators, as well as Ref. 5.1.9. found that the CR of the stratified charge prechamber engine could be increased by 0.3 to 0.5 ratios compared to a conventional gasoline engine operating at a CR around 6.5. However, the more recent investigators, working at CRs around 8, have not increased the ratio and GM found that at a main chamber mixture strength of 13:1, i.e. reducing the stratification between the cylinders, the octane requirements were the same as for the conventional gasoline engine. This latter conclusion is now widely accepted.

Topic Area (b) Combustion Characteristics

The various publications on the Honda CVCC process imply that the mixture in the main combustion chamber burns very slowly to give a high exhaust temperature. Both Ford and Nissan disagree with this observation. The Nissan paper states that the overall combustion rate of the prechamber engine is similar to that of the conventional gasoline engine at lean mixtures, but that the prechamber combustion is faster at rich mixtures. The Ford paper states that main chamber burn time of the prechamber engine is only 5 x crank degrees longer than the conventional gasoline engine. Indeed, the Honda claim to very slow burning must be viewed with reservations, as this would certainly involve a very large penalty in fuel consumption.

Ignition timing of the prechamber engines at mbt (minimum spark advance for best torque) is closer to tdc than for conventional engines, particularly at lean mixtures. This is due to the considerable reduction in delay time between spark firing and the establishment of a flame front. Ford, Nissan and Honda all quote lag times up to 20° later than for the conventional engine.

As with all prechamber engines, the orifice size is the predominant factor influencing combustion speed. The maximum gas speed in the prechamber orifice has been plotted in Fig. 5.6. for various scavenged prechamber engines. For values of $\frac{\text{prechamber volume}}{\text{throat area}} = 0.1$ m the gas velocities are similar to the conventional engine, as recorded by ref. G16 with hot wire anemometry. When Nissan and GM reduced the throat velocity (point 1 of Nissan results, 2 of GM results) the burn rate increased significantly, and combustion cyclic irregularity was reduced. This also involved a reduction in exhaust gas temperature. In addition, there was a significant pressure difference generated between the combustion chambers, as the prechamber charge began to burn. This was also observed on the Walker Stratofire prechamber where the instantaneous pressure difference between the chambers was 13 bar.

If the ratio of prechamber volume to throat area is in the region of 0.1 m then the main chamber cylinder pressure diagram resembles that of the conventional gasoline engine, and the maximum pressure difference between the chambers is in the region of 0.1 bar.

Topic Area (c) Performance

Generally there has always been a small loss in specific torque when a conventional gasoline engine has been converted to scavenged prechamber operation. In the case of the GM 5.75 litre engine, the loss was small, around 0.07 - 0.2 bar, and the maximum specific torque was 7.8 bar. Ford estimate that the loss in torque of a scavenged prechamber engine was 0.5 bar leading to an overall loss in peak output of 10%. The maximum specific torque of the Honda CVCC engine was 8.4 bar measured by SWRI, but other authorities have recorded different figures for this engine. A number of European Companies connected with the automotive industry have obtained Honda CVCC cars from Japan. In every case, the maximum specific torque has been in the region of 7.85 bar. The difference in maximum output has been even greater, SWRI recording 47 kW at 5500 rpm, while the European engines gave 37 kW at 4600 rpm. The vehicle tested at SWRI was a demonstration model supplied by Honda. A detailed study of the operating parameters reveals a considerable difference in mixture strength. The overall air fuel ratio at the point of maximum power was 12:1 for the SWRI Engine and 16:1 for the European engine, and this would explain the difference in power. The reason for the different settings has been further explored under topic area (e). The shape of the torque curve for scavenged prechamber engines is similar to that of conventional gasoline engines. Fig. 5.7.

The scavenged prechamber engines will probably always have a lower maximum specific torque and power output than the conventional gasoline engine, although the magnitude of the difference will vary, depending on the intended engine application. If the stratification at full load is reduced, by inducing a mixture of around 13:1 through the main inlet valve and reducing the prechamber air flow, then the volumetric efficiency will be the same as the conventional engine. However, the charge agitation during combustion will increase the energy lost in heat transfer and reduce the energy released as work and thus the maximum torque. This probably corresponds to the case of the GM 5.75 litre engine. In contrast, the engine designer may wish to retain the charge stratification between the chambers at full load to reduce nitrogen oxide formation. Therefore a depression of about 0.035 bar must be maintained in the main inlet tract compared to the prechamber inlet tract so that a sufficient "number of prechamber fills" is induced through the prechamber. This, inevitably, will reduce the engine volumetric efficiency and performance by about 0.5 to 0.8 bar BMEP.

Road tests conducted on Honda CVCC vehicles gave the following results:-

0 - 60 mph

1.5 litre Honda CVCC (37 kW engine, tested by Ricardo)	17.8 seconds
1.5 litre Honda CVCC (probably 47 kW, tested in Japan)	15.2 "
1.2 litre Honda civic Automatic transmission	18.5 "
1.2 litre Datsun 120 Y	15.8 "

Some of the Russian scavenged prechamber engines gave improvements in maximum output compared to the conventional counterparts, but this was probably a direct result of the increased compression ratio.

Topic Area (d) Fuel Consumption

The references appeared to be in disagreement as to the potential of the scavenged prechamber engine with regard to fuel consumption.

Test Bed Results.

The single cylinder investigation carried out by Ford, indicated that the optimum fuel consumption of the scavenged prechamber engine was 3% worse than the conventional engine. In addition, this point of maximum economy occurred at an air fuel ratio of 19.3 rather than 17 as for the conventional engine. However, Ford also found that when the conventional and scavenged prechamber engines were adjusted to reduce NO_x emissions by E.G.R. and retarded timing, the scavenged prechamber engine had an advantage in fuel consumption.

In the single cylinder work carried out by Wimmer and Lee, ref. 5.1.9. the fuel consumption comparison between scavenged prechamber and conventional engines was found to depend on engine load. At low loads the scavenged prechamber engine had an advantage over the conventional engine while at high loads the position was reversed, fig. 5.12.

Results from 4 cylinder scavenged prechamber engines from Honda (1.5 l), Eaton (2.0 l) and British Leyland (1.8 l) are plotted in Figs. 5.8 to 5.11. The method of presentation, i.e. extrapolating the results to a larger engine is imperfect, as these small engines will operate at a higher speed than if they were constructed to a larger size. However, the trends are still obvious, indicating that consumption is slightly worse than the baseline diesel at low speed, but considerably better at high speeds. The results also improve on those of the baseline conventional gasoline engine presented in the baseline section.

Vehicle Results.

The steady state economy of the GM Impala scavenged prechamber engine was lower than that of the stock vehicle. However, in the CVS tests the results were relatively good. In fact, both the GM and Ford vehicles with inertia weights above 1800 kg gave fuel economy results close to the baseline graph for gasoline engines, Figs. 5.13 and 5.16. Earlier GM results gave low economy, Fig. 5.14.

In contrast, the results from the Honda CVCC car have usually been lower than on the baseline gasoline engine at 2-4 g/mile NO_x. Figs. 5.16, 5.17 and 5.19. Only one recent result, Fig. 5.17, from a research engine at Honda has equalled the economy of the gasoline engine, but the emissions were outside the primary target.

The economy of the Ford 1350 kg vehicle was particularly good at high HC levels, but the economy deteriorated as the emissions were reduced. Fig. 5.15. At an HC level of 1.3 g/mile the fuel economy was better than the conventional engine. However, to reach an HC level of 0.55 g/mile the ignition timing was retarded to increase the exhaust reactor temperature, and this reduced the fuel economy by 17%. Honda carried out a similar exercise on a 900 kg car and achieved the primary emission target with a 10% loss in fuel economy. Fig. 5.17.

The test bed and vehicle fuel economy figures for the Honda CVCC car contradicted each other, the test bed data indicating good economy, while the vehicle data indicates poor economy. This cannot be explained.

The secondary emission target could also be met using E.G.R. Figs. 5.14 and 5.17, as shown by GM and Honda. The Honda engine had a very poor fuel economy, but in the case of the GM engine the economy penalty was partially recovered by increasing the back axle ratio and using lock-up on the automatic transmission. Driveability on this car was exceptionally poor. Fig. 5.20 shows the relationship between fuel economy and NO_x for the GM engine, using the standard automatic gear box.

Topic Area (e) Exhaust Emissions

Lean Reactors.

The HC emissions generated in the combustion chamber of the scavenged prechamber engine will not be significantly different to those of the conventional engine. Since the prechamber engine will operate at a leaner mixture strength there is sufficient oxygen to complete the combustiin of the HC's in the exhaust. There is an optimum air fuel ratio for HC oxidation since the exhaust temperature decreases as the air fuel ratio is increased. The beauty of the scavenged prechamber engine is that this optimum air/fuel ratio for HC oxidation can be maintained to very low loads, while the conventional gasoline engine must be enriched at this point to avoid poor driveability. To give the best possible conditions for exhaust oxidation, the exhaust gas should reside in a high temperature zone for as long as possible. A thermal reactor is used for this purpose. Ref. 5.7.2. has studied HC oxidation with gas residence time and temperature. HC reaction increased asymptotically to a 100% value with increasing residence time, but the 'knee' of the curve occurs at a reactor volume corresponding to 1½ times engine swept volume. However, this figure is only a rough guide as the HC oxidation also depends on:

(a) The mass flow as well as the residence time.

(b) Fluid dynamic state of the flow, i.e. turbulent or laminar.

(c) The previous history of the gas temperature and flow within the reactor.

Carbon monoxide emissions may also be partially controlled by a thermal reactor, although the reaction rate is slower than for HC. Indeed, under certain conditions, carbon monoxide may also be generated within the reactor, due to partial oxidation of HC.

It is important to remember that the scavenged prechamber engine and thermal reactor must be considered as an integral unit for HC and CO emission control. Thus, single cylinder HC emissions cannot simulate the level of the multi cylinder counterpart.

Test Bed Results HC.

There is general agreement among the references that the minimum HC level occurs at an air/fuel ratio of around 17, Table 5.1. In single cylinder tests, the actual level of the scavenged prechamber engine HC's was found to be either similar to that of the conventional gasoline engine (by Nissan) or higher (by Ford). The actual baseline HC levels recorded from the convention engine, by the two companies, were different. When ignition timing was retarded, the HC level of both engines decreased. The actual level of HC emissions at mbt timings for various single cylinder investigations were:

GM	0.37 g/ihp hr.	
VW	4.5 g/bhp hr.	about 2.4 g/ihp hr.
Nissan	1.5 g/ihp hr	
Ford	1.3 g/ihp hr.	

By contrast, the 4 cylinder Honda CVCC engine which contained a thermal reactor returned levels below 0.1 g/ihp hr.

Emission figures taken from 1.5 litre Honda CVCC engines are shows in Figs. 5.8 and 5.9, one covering the load range at 2 speeds and the other the road load. In both cases the HC levels are very low, except at full load where the mixture strength of the SWRI engine was enriched, and therefore HC levels were high as there was insufficient oxygen for oxidation. The British Leyland engine results were slightly higher than the current good gasoline engine, but the engine did not have a thermal reactor. Eaton Corporation obtained good results using a lean reactor and intake swirl. One factor not revealed by these tests is the high HC emissions during idle. Apparently the thermal reactor is ineffective in this mode. Levels of 600 ppm and 300 ppm (C) have been recorded compared with 200 ppm for many conventional gasoline engines.

Results from the Heintz engine were poor, with minimum levels around 4 g/bhp hr,HC.

Test Bed Results NO_x.

NO_x level depends principally on the total air fuel ratio and the degree of stratification as well as the engine load. Since the stratification is influenced by the number of prechamber fills and the shape of the torch opening, one might expect wide variations in NO_x levels between different investigations. This is confirmed below where six NO_x emissions are tabulated at an air/fuel ratio of 20, at a mid-load condition.

GM	4.5 g/ihp hr.	
VW	14.1 g/bhp hr.	about 8 g/ihp hr.
Honda	0.9 g/ihp hr.	
Ford	8.0 g/ihp hr.	
Nissan	0.3 ihp hr.	

These levels may either be higher or considerably lower than a conventional gasoline engine if the conventional gasoline ergine is operating at mixture strengths leaner than 16:1.

The leaner the total air/fuel ratio of the scavenged prechamber engine, the lower will be the NO_x emissions, but usually a trade-off must be made with rising HC emissions above an air/fuel ratio of 17. Honda results also indicate that smaller prechamber throats will reduce NO_x.

If NO_x levels are to be reduced below the primary target, then E.G.R. can be used. According to Ford, introducing the E.G.R. through the prechamber valve compared to the main valve can give lower HC emissions for the same NO_x level achieved. In addition, there is a distinct optimum NO_x level at about 1.5 g/ihp hr below which the scavenged prechamber engine is more efficient than the conventional engine with E.G.R.

Vehicle Emission Results.

A series of published results from scavenged prechamber engines is shown in Figs. 5.13 to 5.19. The GM Impala could not meet the primary emissions target without a thermal reactor, Fig. 5.13. Even when a thermal reactor was fitted the CO level was only marginally within the target. Ford experienced even greater difficulty in meeting these targets with a 2.3 l scavenged prechamber engine in a 1350 kg car. However, in another car a special V8 cylinder head design incorporating uni-sided ports and an inboard thermal reactor gave exceptionally good results, with HC and CO levels well within the target as well as good fuel economy, Fig. 5.18. This particular result was recorded after 30,000 miles of durability testing. In general, the durability aspect of the thermal reactor system is completely satisfactory in contast to catalystic devices on conventional engines. After a 50,000 mile durability test on a Honda CVCC car the emissions were 0.18 g/m HC 2.12 g/mile CO and 0.89 g/mile NO_x.

The European Honda CVCC car tested by Ricardo was within the primary target. It was explained in topic area (c) that this engine had low performance, due to the retention of stratification and resulting high air/fuel ratio at full load. This high air/fuel ratio results in low HC and CO under these conditions. If the mixture is enriched to 13:1 at full load, then HC and CO rise dramatically but the full load torque is also improved. Therefore, there are two distinct trade-off areas, one between HC and CO emission with economy (Fig. 5.13 and 5.15) and the other between HC and CO emissions and performance. However, the second compromise is probably not strictly necessary. When richer mixtures are used at full load, the resulting HC and CO emissions could be oxidised in the thermal reactor if air were added from an air pump. No-one has yet explored this possibility, but the extra production cost of the air pump must be considered.

Honda have achieved the secondary emission target but at a considerable fuel economy penalty and, according to a press statement, a 20-25% loss in power output. The GM Impala also achieved 0.4 grms/mile NO_x but at the expense of HC and CO emissions. (Fig. 5.14).

The CVCC concept employs some extra components in comparison to the conventional gasoline engine, principally to control HC emissions. The most important examples are the lean thermal reactor, the throttle dashpot and positioner and the cold start advance. The throttle dashpot on the Honda CVCC car prevents high manifold vacuums and the resulting disturbance of the mixture strength as liquid fuel evaporates at the low pressures. As shown in Fig. 5.19 the removal of the throttle dashpot and positioner destroys the stratification settings and increases the emissions. The inclusion of low temperature spark advance also has a strong influence on the emissions. The effect of the lean reactor cannot be completely separated, due to the requirement for intake heating. However, Ford have shown that the removal of a reactor liner increases HC from 0.29 to 0.38 g/mile and CO from 2.83 to 4.32 g/mile.

H2

The retrofit systems incorporating automatic valves all have exceptionally high HC emissions. One example manufactured by Morghan using a similar principle to that shown in Fig. 5.4. was tested on a vehicle and gave 14 g/mile HC.

Other Emissions.

Since the scavenged prechamber engine introduces two carburetted mixtures, there are no rich regions at the time of combustion. The rich prechamber mixture is diluted with the main charge, so smoke would not be generated during combustion. Tests conducted on the Honda CVCC car verify that the smoke emissions are very low. Fig. 2.28. Odour levels are also relatively low, Fig. 2.29. The unregulated emissions such as particulates and ammonia are at a similar level to the conventional engine, while aldehyde emissions are exceptionally low, Figs. 5.21 - 5.26. The latter conclusion is surprising since 'lean burn' engines usually have high aldehyde levels. The thermal reactor may help in the removal of aldehydes.

Topic Area (f) Multi-Fuel Capability

The carburetted scavenged prechamber engine is unlikely to have a multi-fuel capability. When fitted with port fuel injection, the Heintz engine was able to operate on 68 RON, JP4 and diesel fuel. However, emissions would probably be poor with fuel other than gasoline, and the engine would not start on these alternative fuels.

Topic Area (g) Engine Components

The Honda CVCC has apparently not experienced difficulties during durability testing. This is rather surprising as thermal reactors on conventional engines frequently fail during long term tests. The success of the Honda reactor may result from its simple design, in comparison to the large, complex and more effective types manufactured by Du Pont. According to GM, the high exhaust temperature may also impose requirements for better exhaust valve materials.

The operation of the 3rd valve from a separate cam, rather than the main inlet rocker, will prevent potential problems. The extra inertia of the prechamber valve gear on the main inlet cam could lead to wear and valve bounce in high speed engines. The Russian engines which use this type of design had a limited speed range.

Thermal stress concentrations at the throat frequently occur during the development of diesel prechamber engines, but this problem can also be avoided by using a stainless thimble as the prechamber. This could be a significant problem area with the Heintz design, due to the location of the prechamber between the valves and the multiple orifices.

The carburettor of the 3 valve engine is more complex than the counterpart on the conventional gasoline engine. In addition, the manufacturing tolerances are also tight, as explained in topic area (a).

Topic Area (h) Cost

Honda have made estimates of the extra cost on the 3 valve engine in production, and have stated that it costs between $ 160 to 200 more when fitted in a $ 2000 car, compared to the conventional counterpart. However, the gasoline engine does not possess the same potential for low exhaust emissions without the fitting of an oxidation catalyst and E.G.R.

Estimates for retrofit prechambers with automatic valves in existing engines have varied between 50 and $ 115 for a 4 cylinder engine.

Topic Area (i) Mathematical Models

Ref. G4 has identified a critical point with respect to NO_x formation in stratified charge engines, i.e. the proportion of the mixture which burns at a stoichiometric mixture strength has the greatest influence on the amount of NO_x produced. In other words, it is not important how rich or how lean the main and prechamber mixtures are, but only how much of the charge exists in the mixture cloud near the throat, since the air fuel ratio of this portion is close to the stoichiometric. This, of course, is because the characteristic of NO_x formation against mixture strength comes to a peak at a point just lean of stoichiometric. This observation explains many of the trends observed in this survey. The FCP engine had relatively high NO_x emissions as the region of stoichiometry on the combustion chamber had the form of half a hemisphere, with the injector at the centre, and this proportion of the charge was relatively large. The 3 valve prechamber has a mixture cloud at the throat, but the stoichiometric volume proportionally smaller as it only occurs in a limited region of the chamber. The engines with blank prechambers and fuel introduced to both chambers by injectors gave the lowest NO_x emissions, as there was no mixture cloud. The only stiochiometric region was an interface area at the throat between the rich prechamber and the lean main chamber. This simple explanation cannot be applied to the category 2 and 3 engines since fuel in burning from the surface of droplets or as liquid fuel, and the actual mixture strength is not known.

The mathematical models in the reference can be divided into 3 groups:

(a) Mixing Models.

(b) Combustion and NO_x formation models.

(c) Cycle simulation models.

Mixing Models.

As already explained under topic area (a), the overflow of rich mixture from the prechamber and subsequent formation of a mixture cloud during the induction stroke is important to the combustion. However, this actual mixing mechanism has not been predicted, since its shape and surface finish of the orifice will exert a large effect. GM and Honda have overcome this problem by sampling in their respective engines to determine the amount of mixing which occurs. The fact that each has come to different conclusions with regard to mixing is indicative of the sensitivity to the engine design. This area would require much more study before the mixing could be predicted from fluid dynamic theory.

Combustion Models.

The combustion models in the references are very crude, and are generally simple homogeneous charge models divided into two regimes. None of the references have attempted to model the torch ignition process which obviously must play an important part in the heat release characteristics. As an example, the GM model assumes that the spark occurs at the prechamber throat and that the flame proceeds to the prechamber and main chamber. The mixture in each chamber burns according to the Wiebe function, ref. G18 and the burnt gases expelled from the prechamber mix completely with the burnt gases of the main chamber. This model, even though it is probably among the references, does not resemble the physical process taking place. Heat transfer within the prechamber has been taken into account in the programme using the Woschni equation ref. G68 but main chamber heat transfer has been ignored.

The GM study identified an unsteady gas flow phenomena which affected the in-cylinder stratification, particularly at low prechamber flow rates or number of prechamber fills.

Cycle Simulation Models.

Honda have developed a cycle simulation model, so that complete cycle exhaust emissions can be predicted from steady state results. Generally, such a procedure cannot be used for conventional gasoline engines, as large excursions in air/fuel ratio take place during accelerations and decelerations and these conditions cannot easily be simulated under steady state conditions. However, the use of the dashpot throttle controller, the absence of acceleration enrichment, and the high degree of intake heating, mean that the Honda engine does not experience air/fuel ratio excursions of the same magnitude during vehicle transients. Therefore, cycle simulation based on steady state results is justified. It is surprising that HC emissions can be predicted accurately, since a hysterisis effect occurs with thermal reactors, i.e. the temperature within the reactor depends partly on the previous gas conditions and the amount of heating of the reactor which occurred.

Therefore, each of the simulation models while having some use has a limited applicability since we do not understand the process occurring within the scavenged prechamber engine in detail. In particular, the formation of the mixture cloud, the manner in which the combustion flame spreads in the combustion chamber, and heat transfer cannot be quantified.

Topic Area (j) Engine Size and Weight

A number of engines have been modified by Honda to the CVCC concept, without significantly affecting the overall dimensions. In some cases, the inclusion of a thermal reactor in place of the original exhaust manifold has increased the overall engine width by 2 - 5 cms.

The more complex carburettor, thermal reactor and complex piping can affect engine weight. The modifications to the GM Impala increased the engine weight by 12 kg.

Topic Area (k) Noise and Vibration

Noise measurements carried out by SWRI on a Honda CVCC car are shown in Figs. 2.31-2.33. The full load acceleration noise was relatively high, particularly when the small bore size of the engine is taken into account, Fig. 2.33. However, on steady state drive-by and idle tests the noise level was close to that expected, although the fan contributed a large noise component during idling. Cylinder pressure diagrams under full load conditions are not available, but these results may indicate that under non-stratified conditions, combustion in the scavenged prechamber engine is relatively harsh. Noise tests have not been conducted on a European Honda CVCC engine where stratification is maintained at full load, but subjective judgement suggests that full load noise levels are no greater than conventional gasoline engines of the same bore size.

Topic Area (l) Cold Starting

The requirement of the scavenged prechamber engine to have a controlled rate of throttle closure has already been mentioned under topic area (e). In addition, the mixture is not enriched during acceleration, so the stratification is not destroyed under transient engine operation. This observation has been noted by both Honda and GM. If these precautions are ignored, then poor HC emissions and driveability can result. The Russian ZIL engine experienced exhaust backfires during decelerations.

The scavenged prechamber engine appears to reach its operating temperature relatively quickly, probably due to the higher heat rejection of the combustion chambers, and also due to the early fuel evaporation system employed on both inlet tracts. However, the EFE (Early fuel evaporation) system gives rise to problems with unstable fuel metering after hot starts on the GM Impala. It is obviously difficult to arrange adequate manifold heat, and also to completely insulate the carburettor, as hot vapour will rise to the carburettor after the engine has stopped.

Patents Ref. 5.4

A number of patents covering scavenged prechamber engines have been recorded since the early 1920's. A good review of these patents is contained in Ref. 5.1.9. and 5.1.16.

Turbocharging

Among the recent literature there are no references to turbocharging of scavenged prechamber engines, and there would not appear to be any significant advantages to such an approach. However, the principle of retarded prechamber filling with a turbocharger, used by Sir Harry Ricardo might bear further study. The problem of transient turbocharger response which occurs with conventional gasoline engines should be avoided.

Application to Motorcycles

At first sight, the scavenged prechamber combustion arrangement would appear to be ideal for motorcycles, in that the mixture could be set lean, and large gains in economy could be obtained. However, the system is not practical in the smaller cylinder sizes, as the prechamber valve and orifice would be difficult to manufacture. Applying the ratios of prechamber throat areas to prechamber volume of the Honda CVCC engine to cylinder sizes of 50 cc and 250 cc gives throat diameters of 3 and 6.7 mm respectively. While the larger throat size might be acceptable, deposition in the throat of the smaller size would probably affect performance. In addition, the arrangement for intake heating of the carburettor would considerably increase the size of the engine installation.

If the carburettor settings of existing motorcycle engines were adjusted for best economy, there would be little advantage for a scavenged prechamber version of the engine. The low exhaust emission potential of this stratified charge engine for motorcycles is unnecessary at the proposed emission levels.

Conclusions

1. The combustion systems which have been identified in this category can be sub-divided into four groups:

   (a) Those developed recently which resemble the Honda CVCC engine.

   (b) Various Russian pre-chamber engines.

   (c) Retrofit pre-chambers for existing conventional engines.

   (d) Miscellaneous engines, including the Heintz.

2. There was some disagreement in the references as to the existence and importance of a mixture cloud in the main chamber.

3. The full load torque and power output of the scavenged pre-chamber engine depends on the stratification arrangement. Two arrangements are possible. The pre-chamber and main chamber may both be enriched to give an overall air fuel ratio of 13:1 at full load, resulting in specific torque levels only marginally below conventional engines. Alternatively, stratification may be maintained at full load with an overall air fuel ratio of 16:1, resulting in about 10% loss in specific torque. Intake charge heating may also contribute to a loss in specific torque.

4. Fuel consumption of the Honda CVCC car was 10% below good gasoline engines at the primary emission targets. However, U.S. automotive manufacturers have obtained better results on larger engines, so that the overall loss in economy is small.

5. If stratification of the fuel is maintained at full load, HC and CO emissions can be within the primary targets. The scavenged pre-chamber engine relies on a thermal reactor to control HC and CO, and the durability of this component is superior to catalysts. The secondary targets can also be achieved but at the expense of fuel economy and durability, unless an oxidising catalyst is used. However, this leads to an expensive engine package.

   Retrofit pre-chambers suffer from high HC emissions.

6. The engines in this category do not have a multi fuel capability.

7. There are no insoluble problems in producing scavenged pre-chamber engines. Cylinder head complexity is increased, and the tolerances on carburettor manufacture are tightened.

8. There will be some increase in engine cost compared to existing engines, although this will be partially compensated by the avoidance of a catalyst.

9. The scavenged pre-chamber engine will not be significantly larger or heavier than a corresponding conventional engine.

10. If stratification is maintained at full load, noise levels of the scavenged pre-chamber engine are similar to the conventional engine. However, if the

engine is enriched at full load, the noise level increases, due to harsh combustion. Idling noise is similar to conventional engines.

11. Precautions must be taken to avoid destroying stratification under transient conditions. This problem is solved by deceleration devices, but at the expense of engine braking.

Table 5.1. SCAVENGED PRECHAMBER ENGINE OPERATION

ENGINE	No. of Cylinders	Cylinder Size cc	Compression Ratio	$\frac{\text{Pre-chamber Volume}}{\text{Clearance Volume}}$	$\frac{\text{Pre-chamber Volume}}{\text{Throat Area}}$ m	No. of Pre-chamber Fills	Pre-chamber Air Fuel Ratio	Overall Air Fuel Ratio	
								Min HC	Min Fuel Consumption
HONDA (1)	4	487	8.0	0.073	0.109	3.7-idle 7.8-50 mph	3	16-18	
(2)	4	375					3-6 full load 4½-8 road load	13½-17½ full load 15-22 road load	
FORD	1	820	8.5	0.089	0.086	6	6	17	19.5
GM (1)	1	818	7.95	0.055	0.091	1	8	17	
(2)	1	818	7.95	0.055	0.820				
NISSAN (1)	1	570	8.5	0.10	0.269	1.5	6	17	
(2)	1	570	8.5	0.10	0.067				
V-W	1	400	8.3	0.12	0.213	4.5	2.9	17.4	21
WIMMER AND LEE	1	612	6.46	0.11		1	3	17	
HEINTZ	4	805	7.7	0.186	0.785				

Figure 5.1. Honda C.V.C.C. engine and Ford scavenged pre-chamber engine.

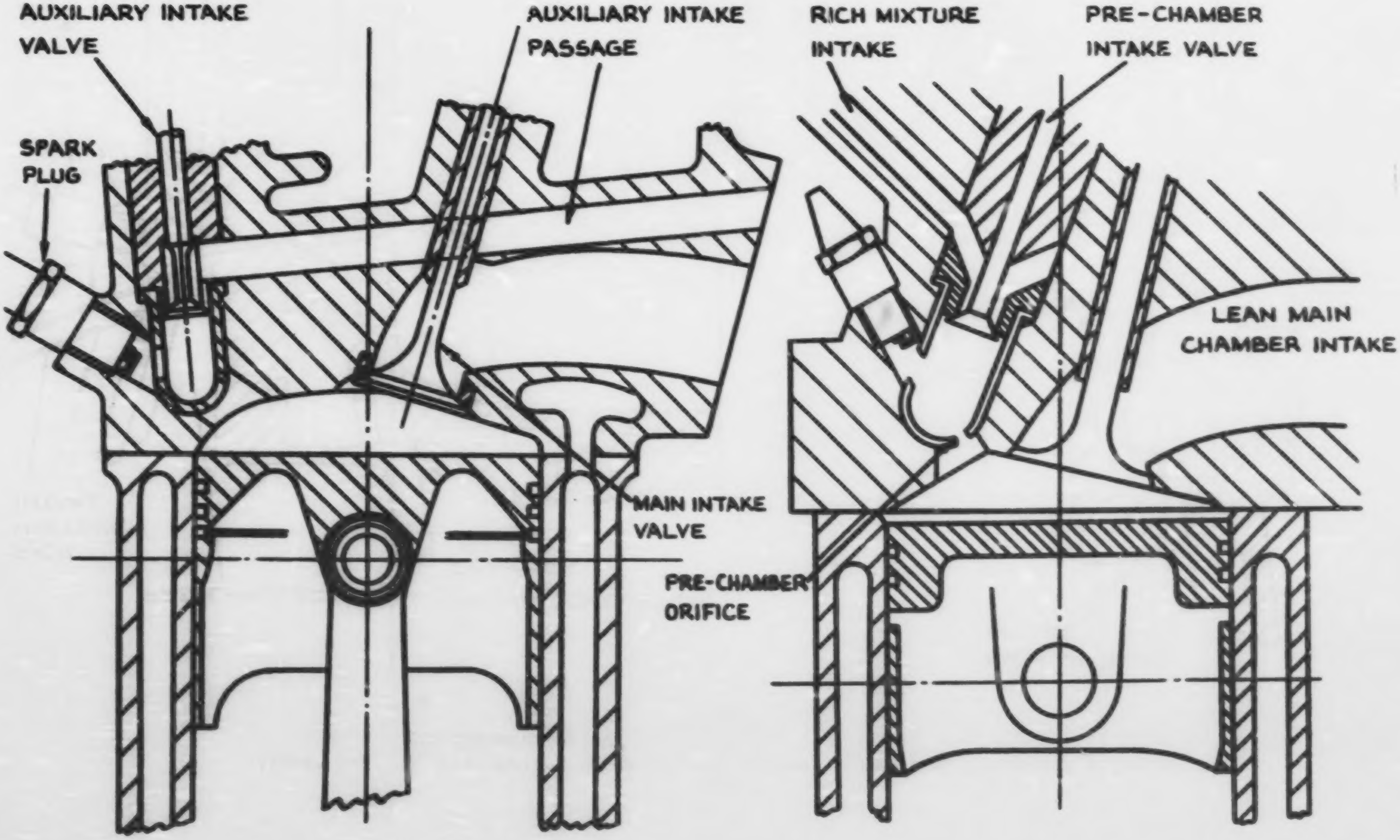

AUXILIARY INTAKE VALVE
SPARK PLUG
AUXILIARY INTAKE PASSAGE
MAIN INTAKE VALVE
PRE-CHAMBER ORIFICE
RICH MIXTURE INTAKE
PRE-CHAMBER INTAKE VALVE
LEAN MAIN CHAMBER INTAKE

Figure 5.2. **General Motors** scavenged pre-chamber engine and Volkswagen scavenged pre-chamber engine.

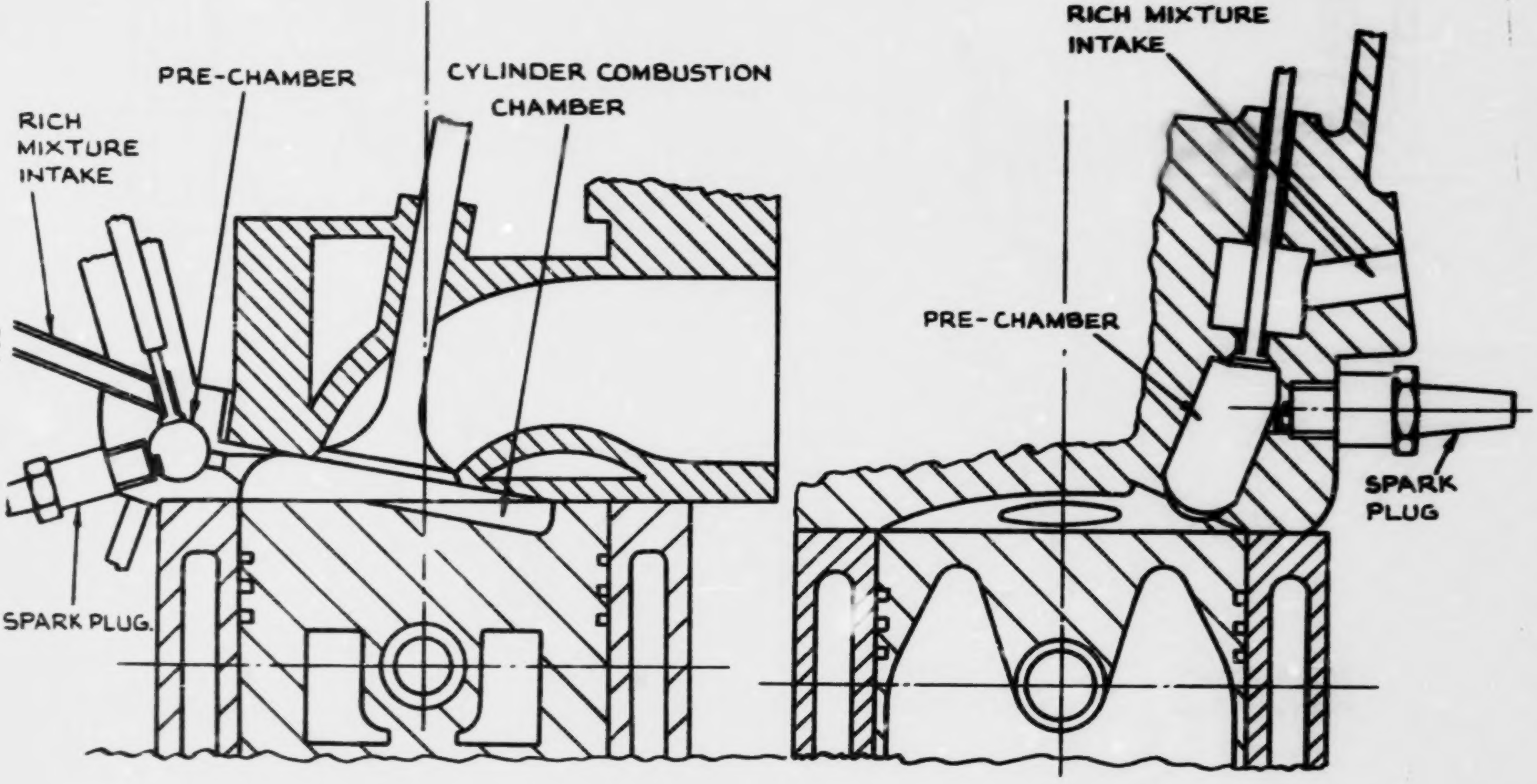

PRE-CHAMBER
CYLINDER COMBUSTION CHAMBER
RICH MIXTURE INTAKE
SPARK PLUG.
RICH MIXTURE INTAKE
PRE-CHAMBER
SPARK PLUG

Figure 5.3. Eaton scavenged pre-chamber engine.

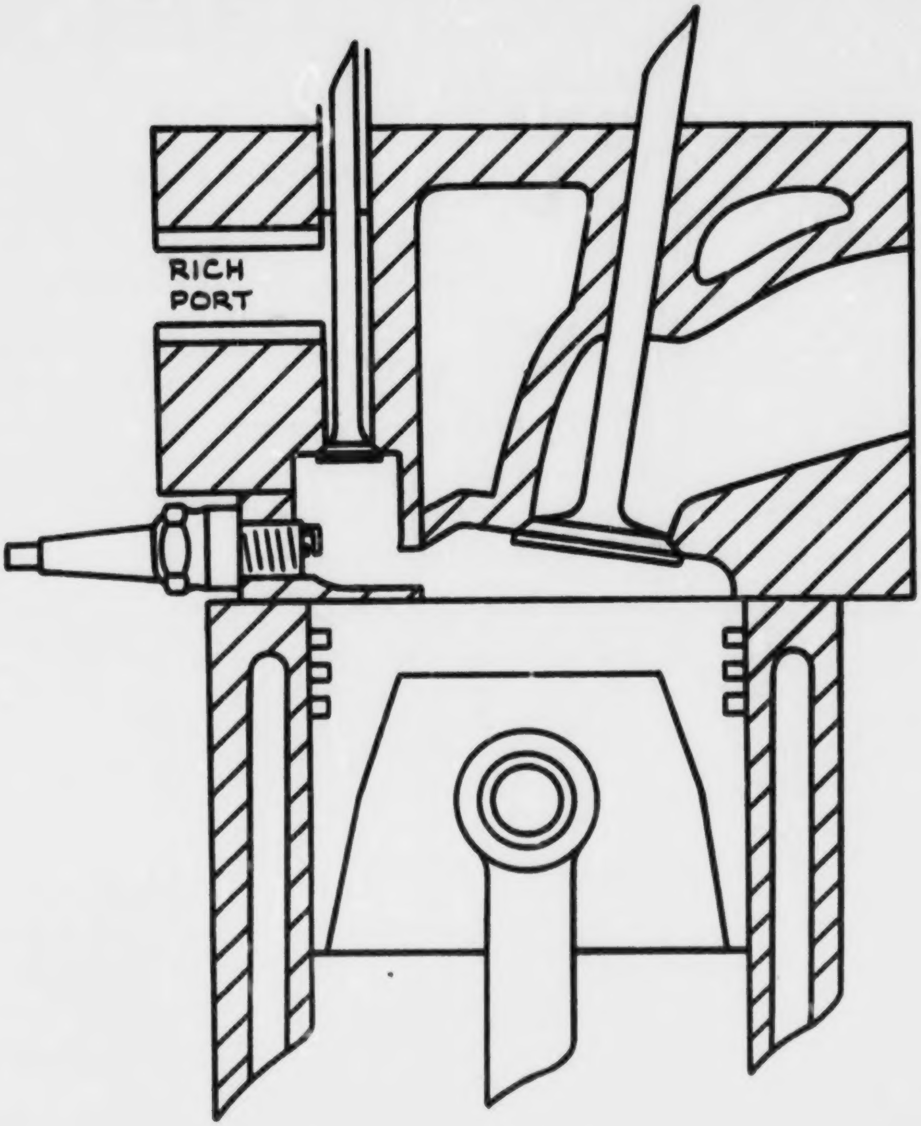


Figure 5.4. Walker stratofire system.

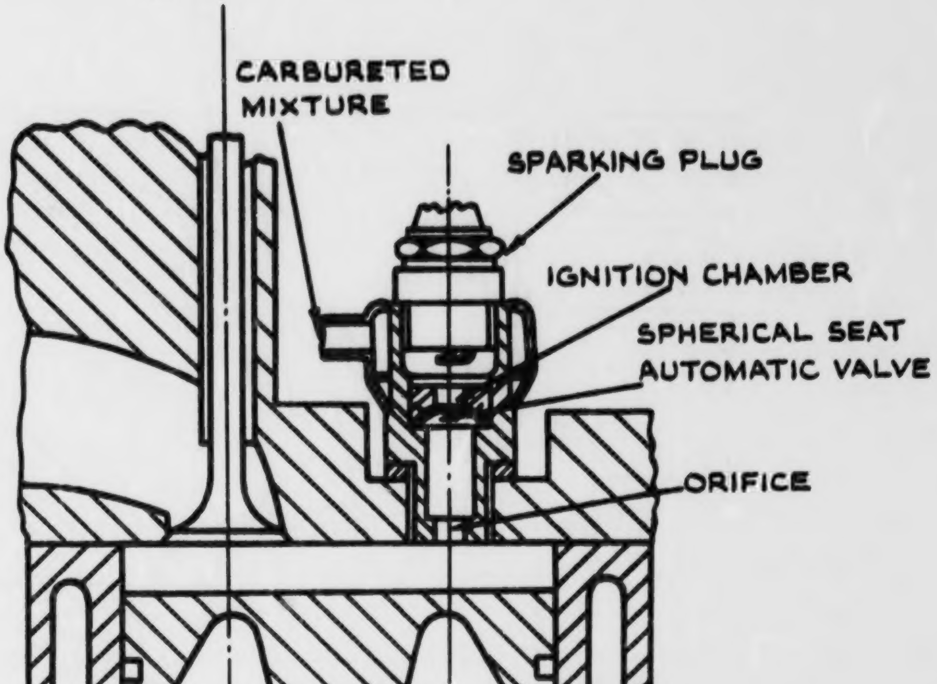

CARBURETED
MIXTURE
SPARKING PLUG
IGNITION CHAMBER
SPHERICAL SEAT
AUTOMATIC VALVE
ORIFICE

Figure 5.5. Heintz.

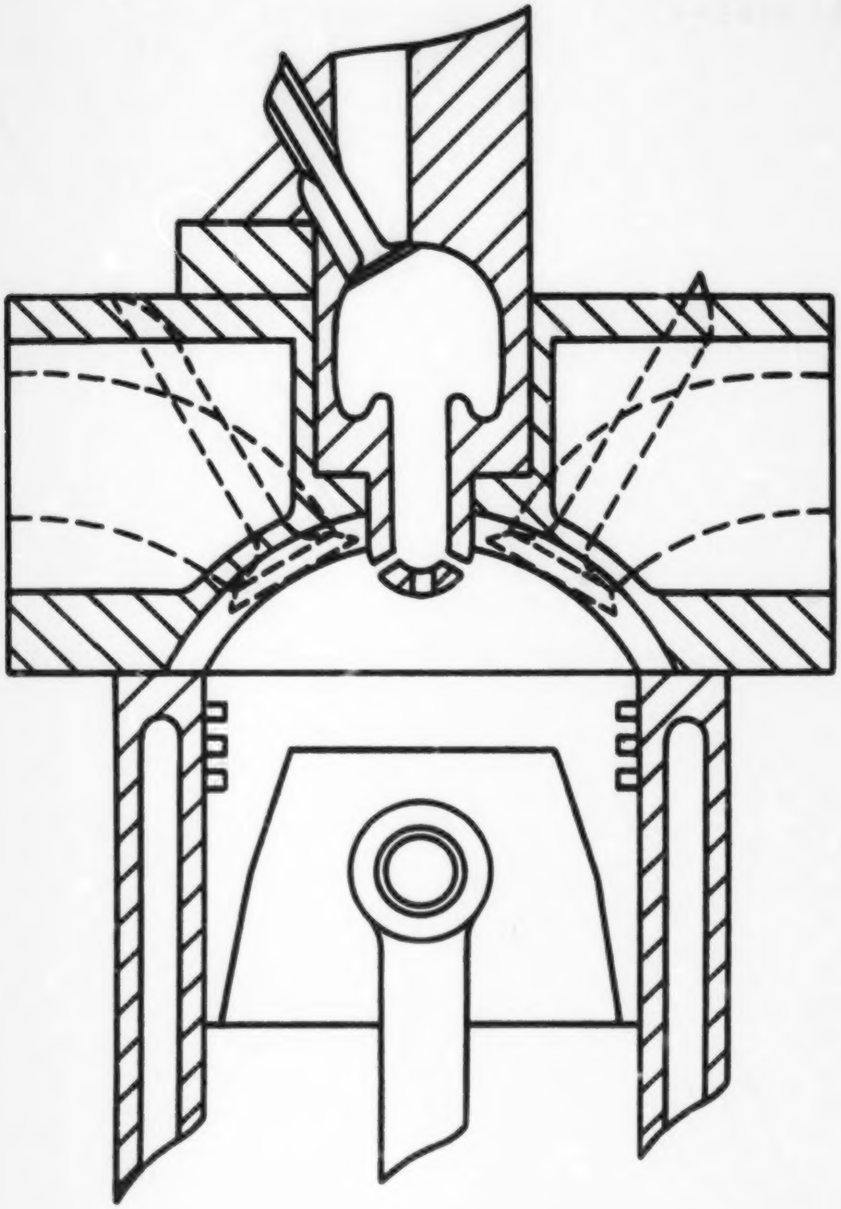


Figure 5.6. Maximum throat velocity during compression for scavenged pre-chamber engines. 1000 rev/min.

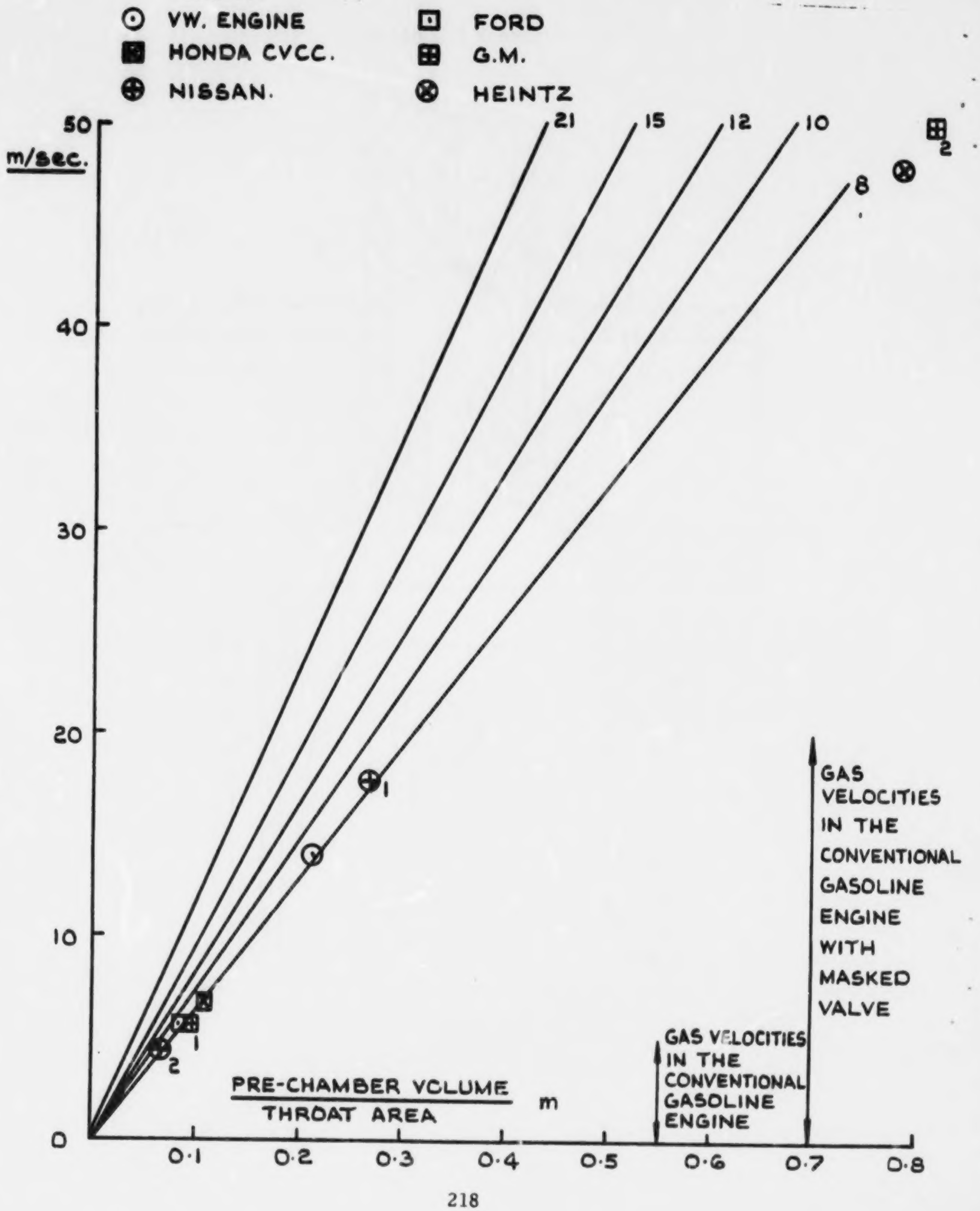


Figure 5.7. Comparison of torque curves. GM pre-chamber engine and baseline diesel.

GM. PRE-CHAMBER ENGINE (DATA SCALED FROM 350 in^3)
SWEPT VOL. FOR 96 kW = 4·75 l (290 in^3) MAX. POWER SPEED = 4000 rev/min.

DIESEL ENGINE.
SWEPT VOL. FOR 96 kW = 4·77 l (291 in^3) MAX. POWER SPEED = 4000 rev/min.

HONDA PRE-CHAMBER ENGINE (DATA SCALED FROM 91 in^3)
SWEPT VOL. FOR 96 kW = 3·8 l (232 in^3).

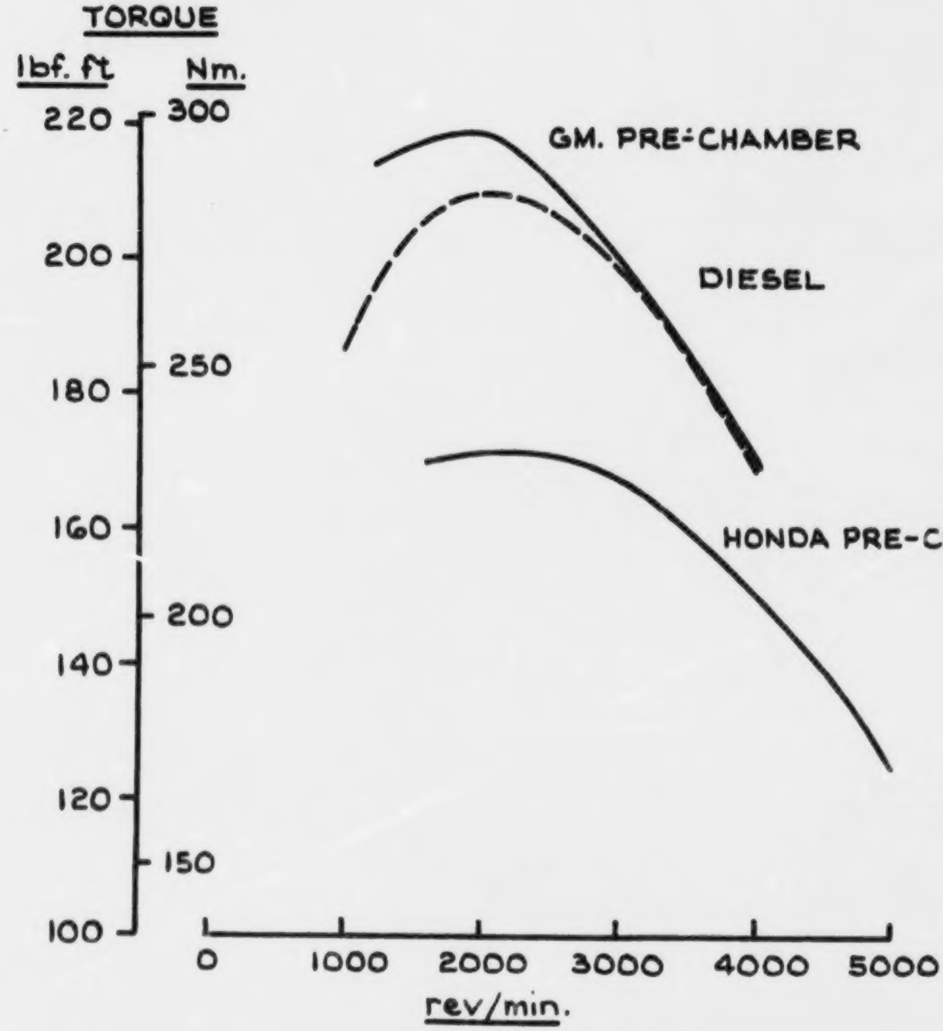


Figure 5.8. Emissions and economy of a Honda scavenged pre-chamber engine results extrapolated from 4-cylinder Honda CVCC engine. (1.5 1) Recorded by SWRI.

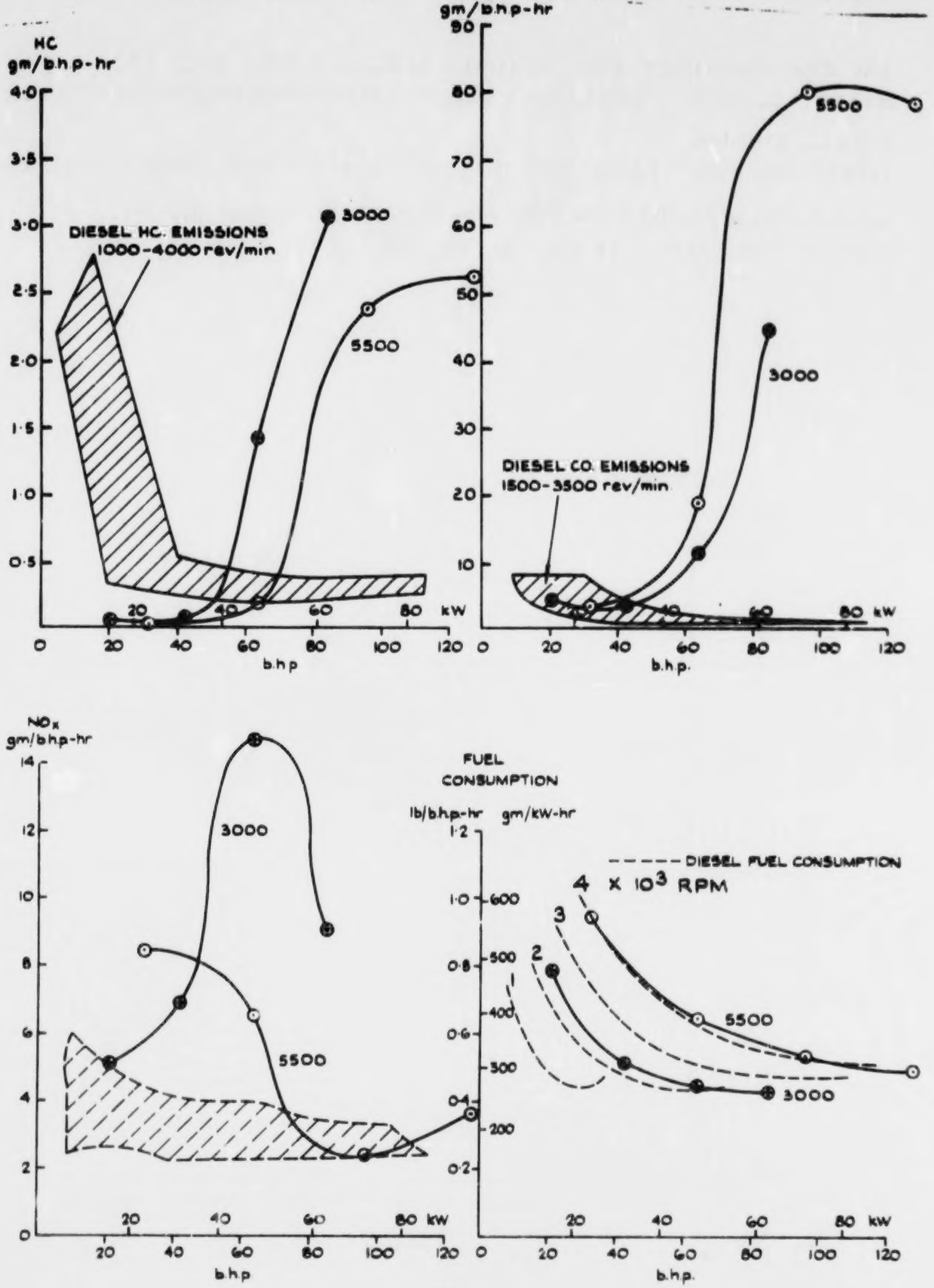


Figure 5.9. Emissions and economy of a Honda scavenged pre-chamber engine results extrapolated from 4-cylinder Honda CVCC engine. (1.5 1)

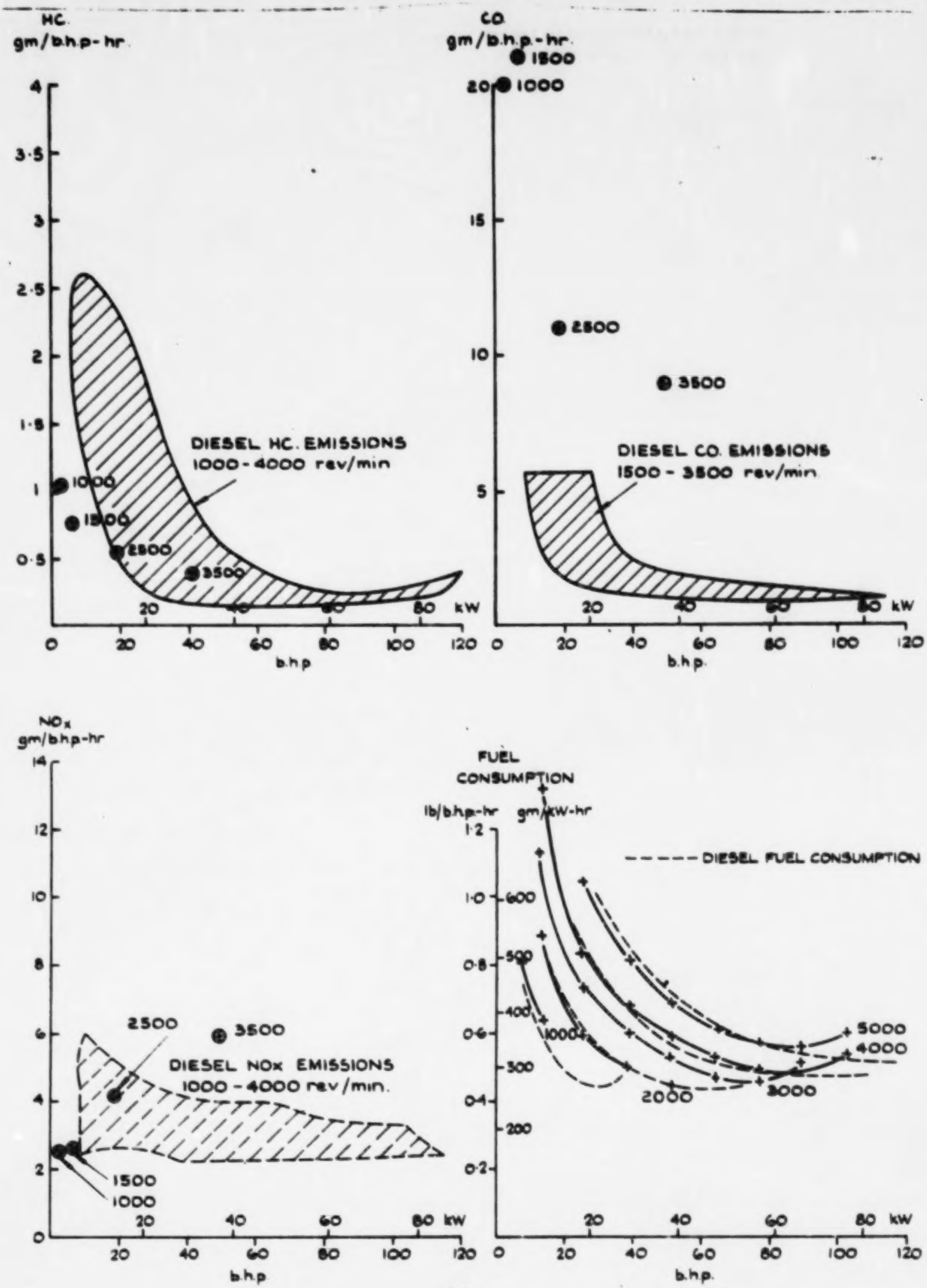


Figure 5.10. Comparison of emissions and fuel consumption of standard and swirl port pre-chamber engines with a Comet Mk V diesel engine. (Eaton) Results extrapolated from 2 l engine.

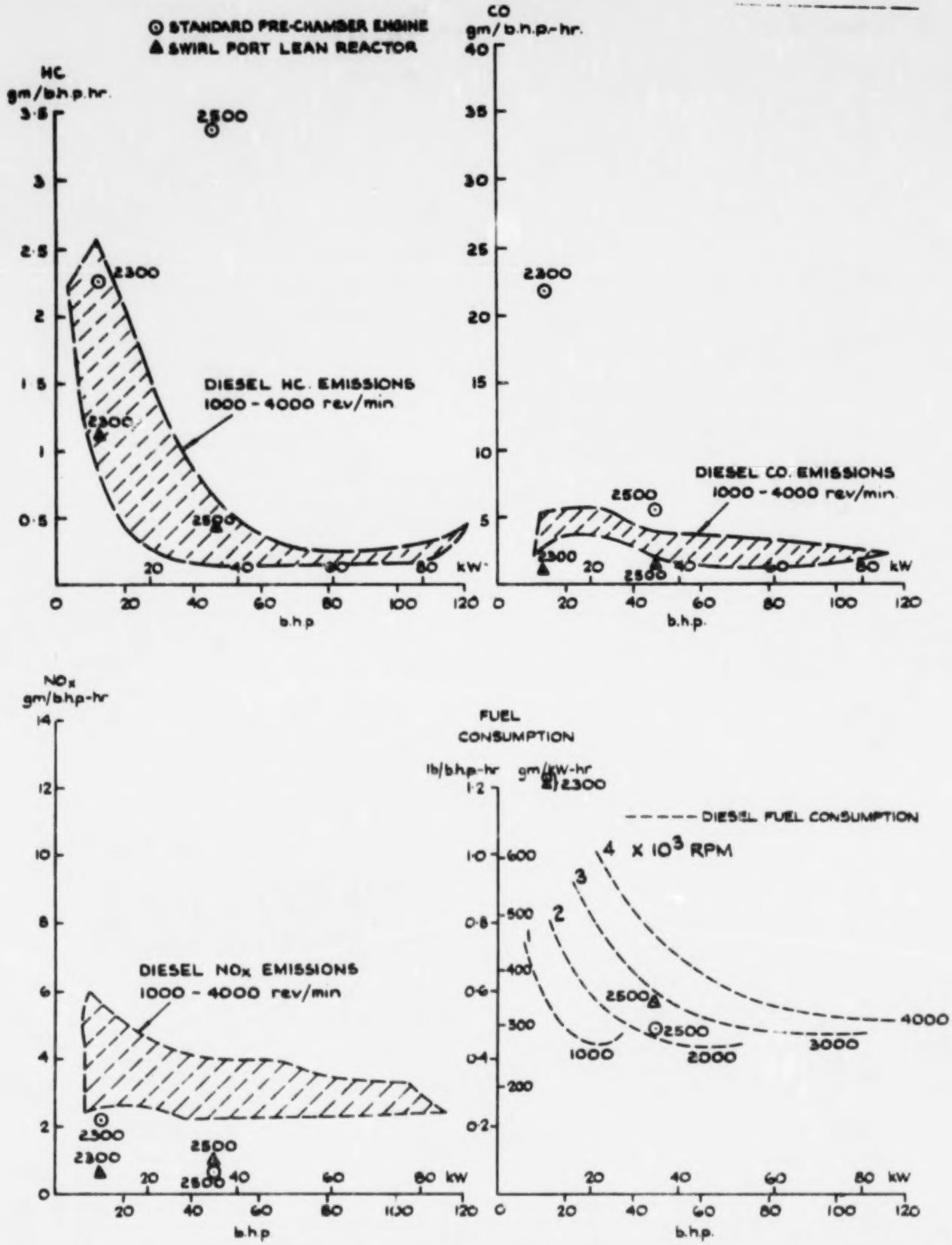


Figure 5.11. Emissions and economy from BLMC scavenged pre-chamber engine results extrapolated from SV I 81 110 in^3.

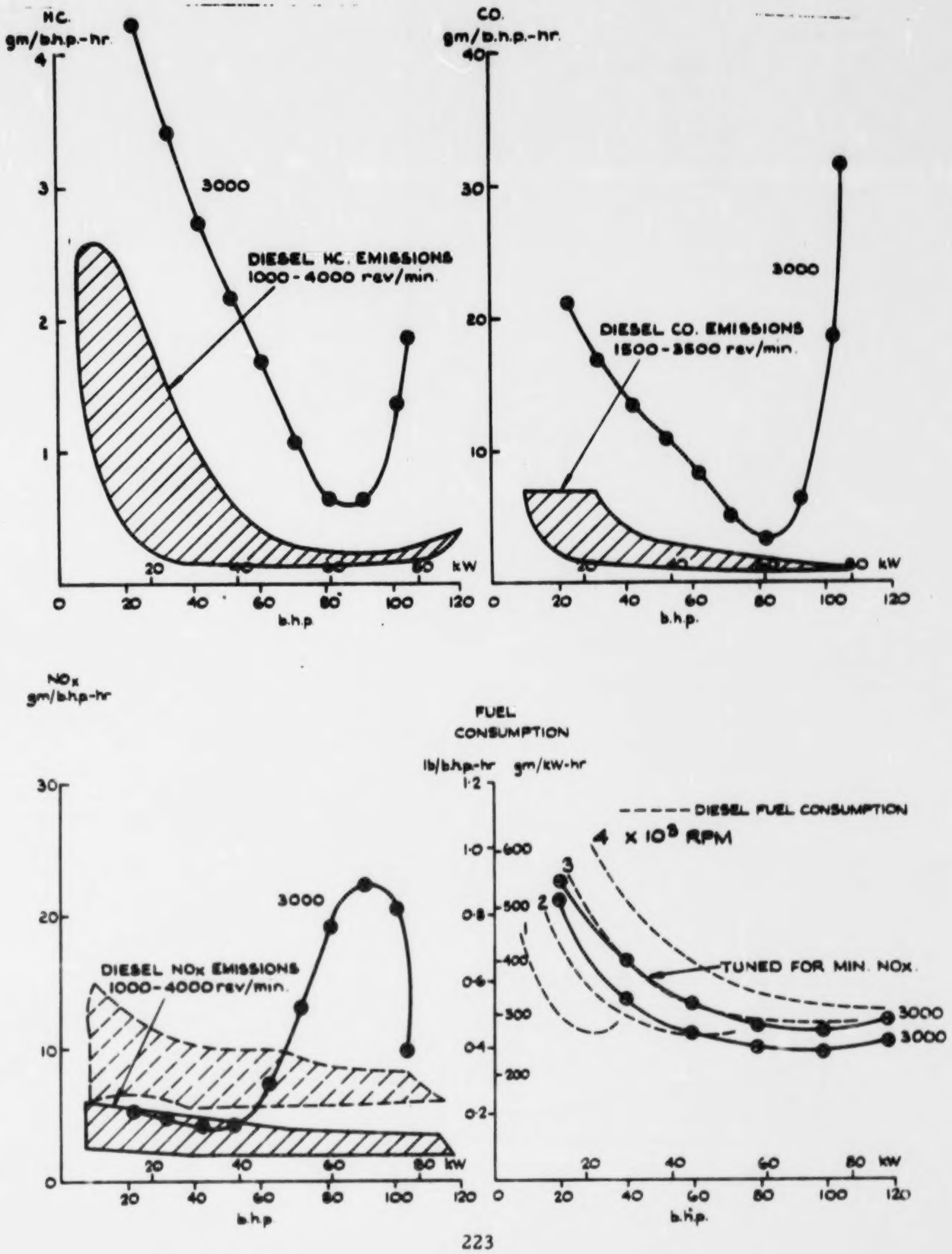


Figure 5.12 Emissions and economy from a scavenged pre-chamber engine, results extrapolated from a single cylinder Wimmer and Lee engine.

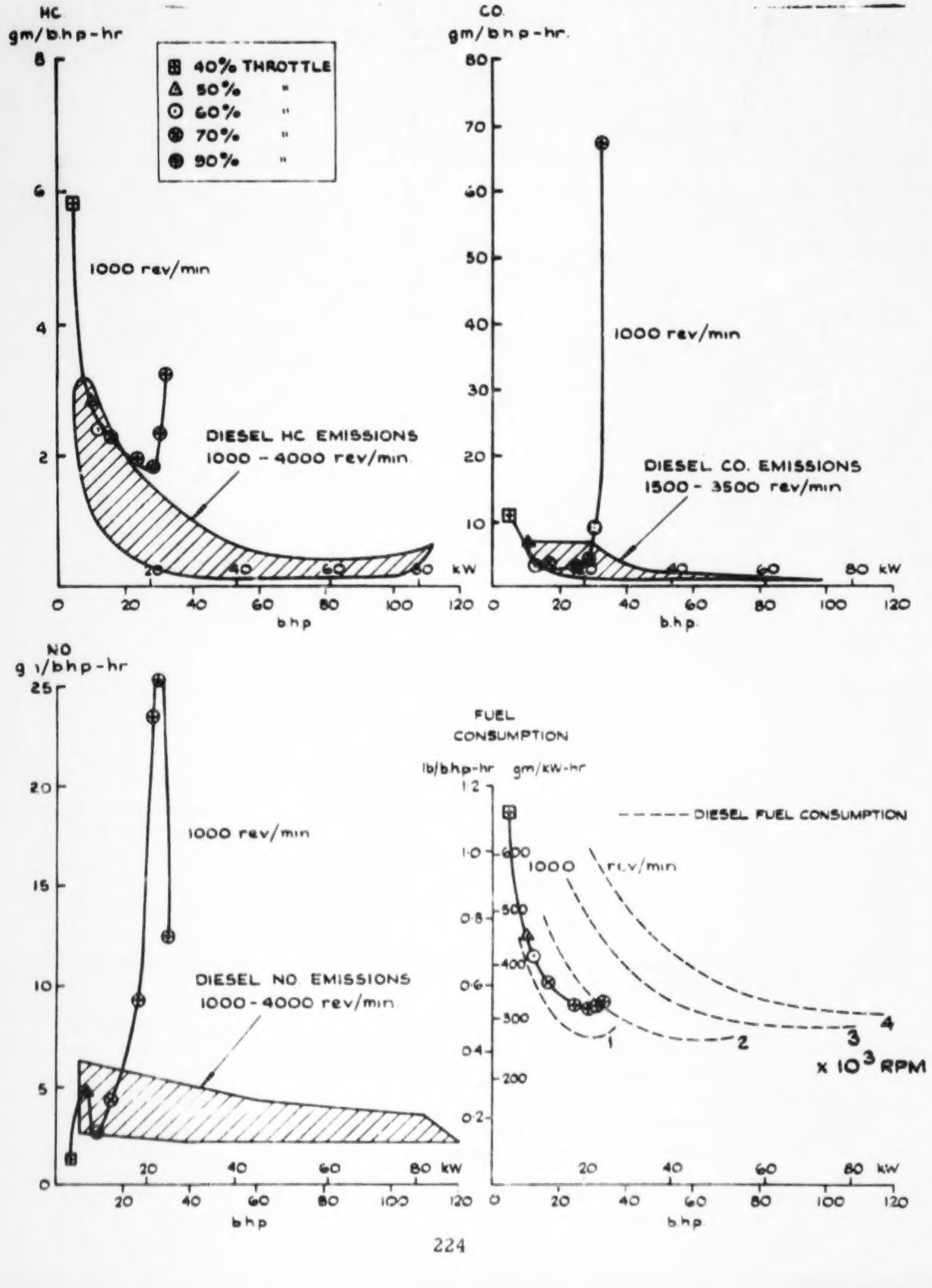


Figure 5.13. Emissions and economy of GM scavenged pre-chamber engine. (5.75 L) Early fuel evaporation and cold spark advance.

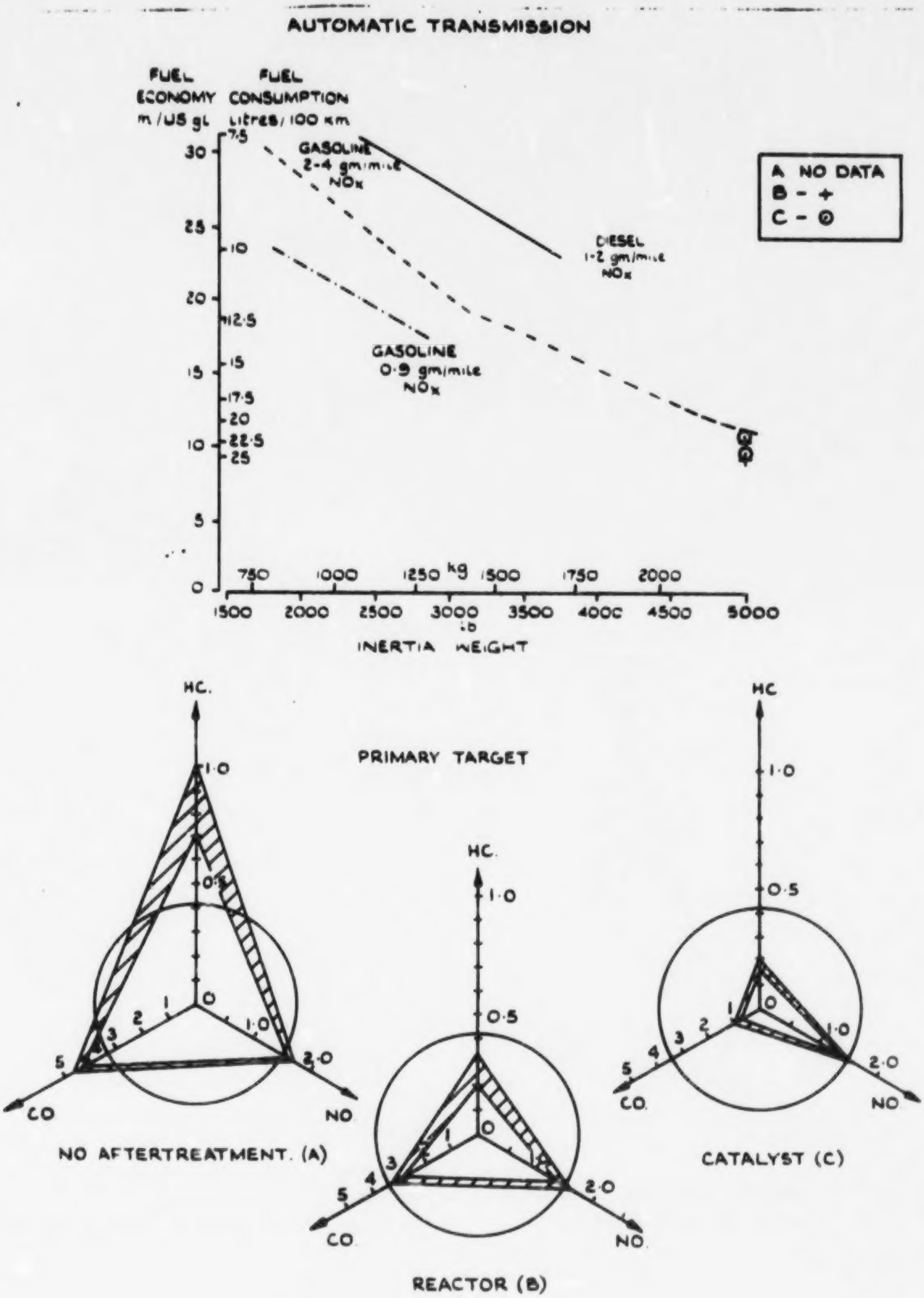
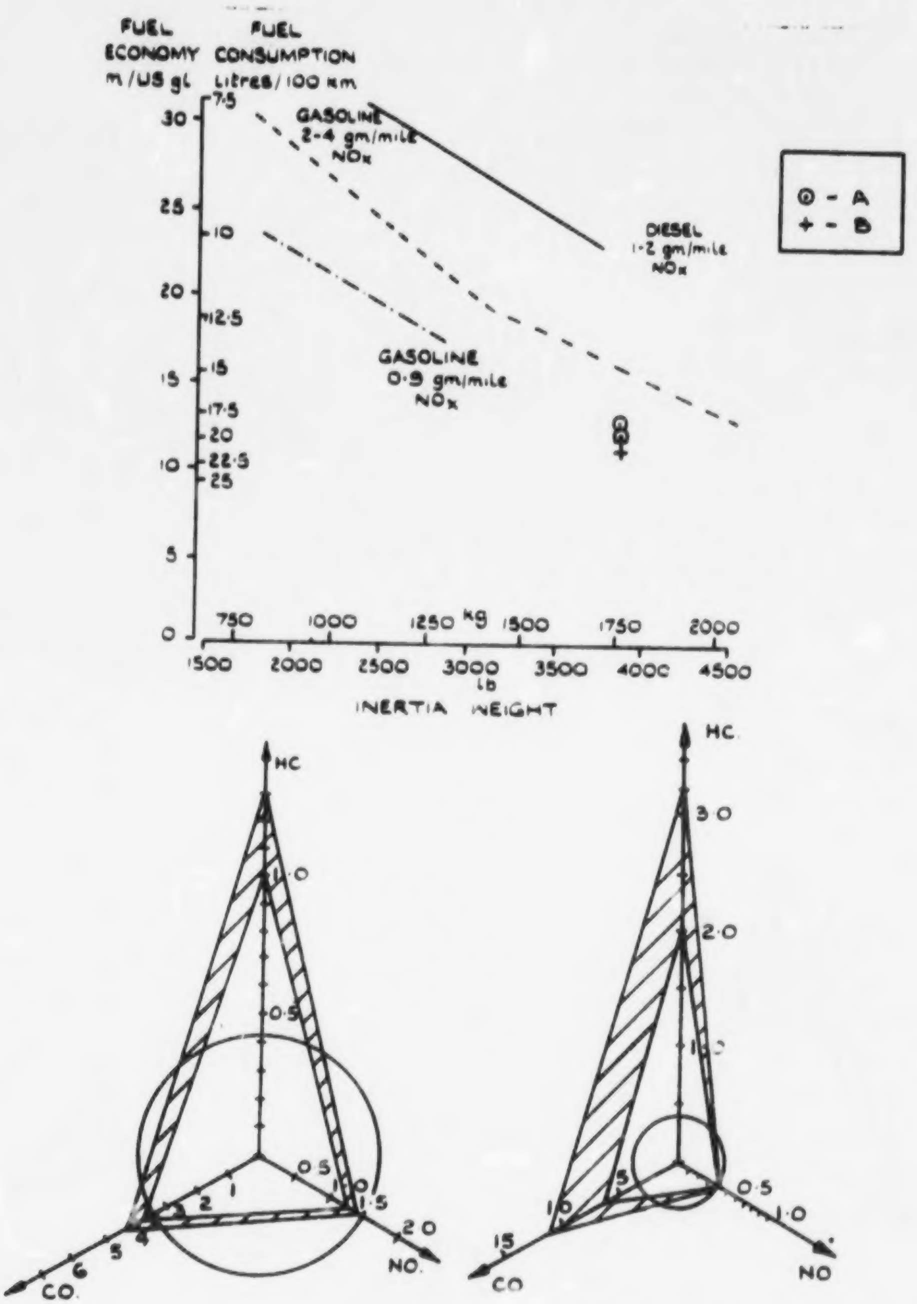


Figure 5.14 Emissions and economy of FM scavenged pre-chamber engine. (5.75L)



PRIMARY EMISSION TARGET (A)
NO EGR 2·73 AXLE RATIO

SECONDARY EMISSION TARGET (B)
EXTERNAL COOLED EGR
3·08 AXLE RATIO LOCK UP TRANSMISSION

Figure 5:15. Emissions and economy of Ford scavenged pre-chamber engine. (2.3L)

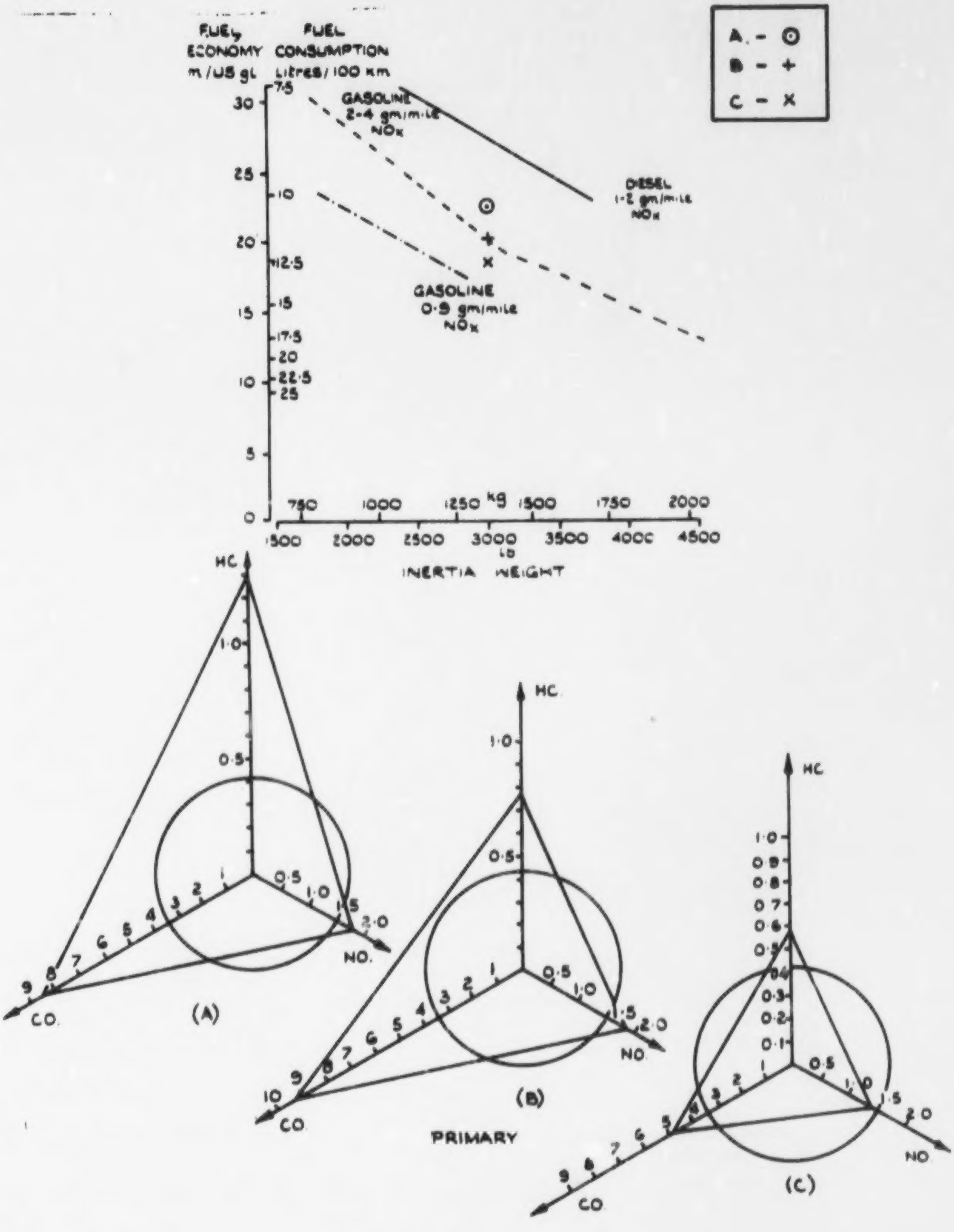


Figure 5.16. Emissions and economy from CVCC vehicles.

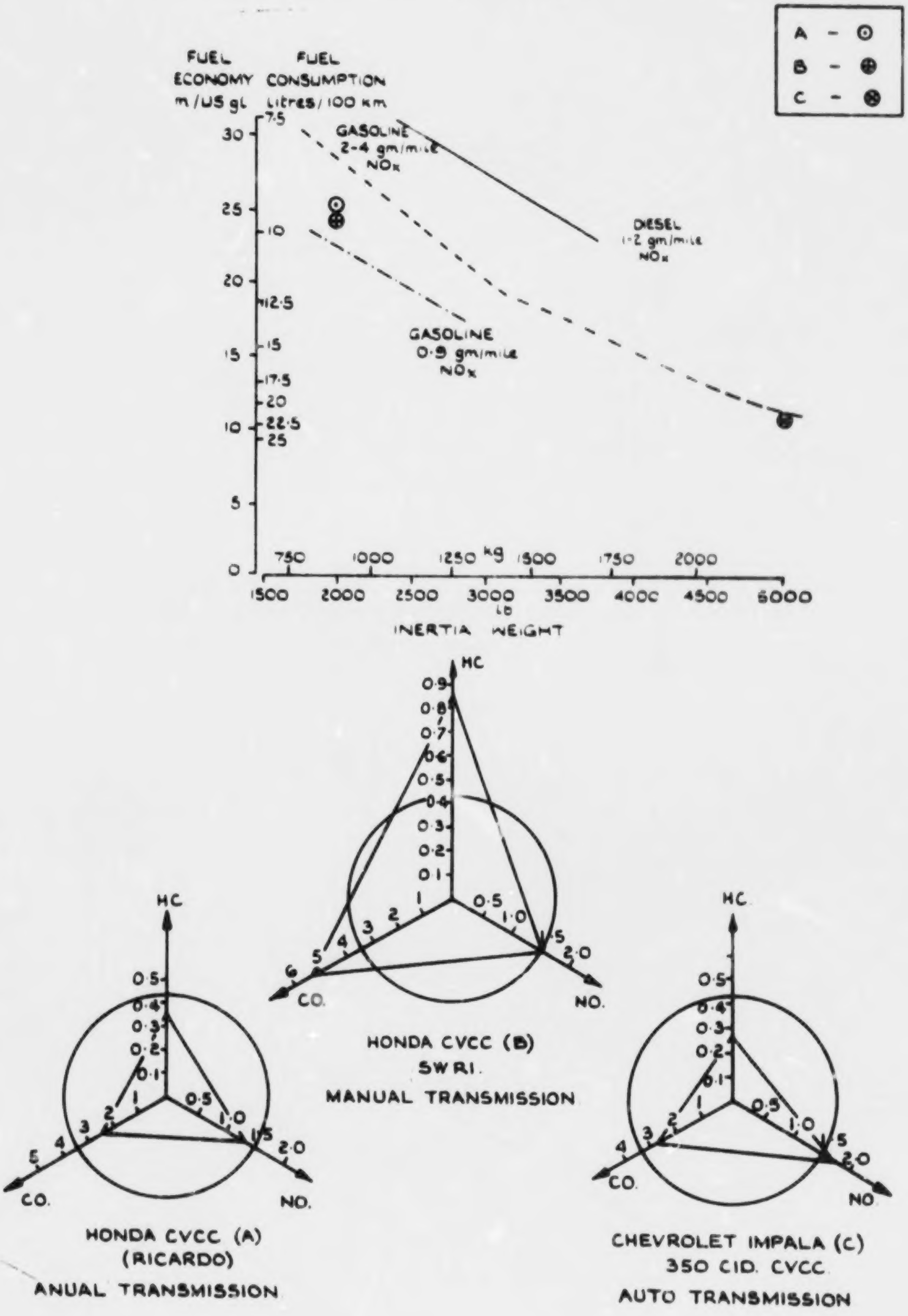


Figure 5.17. Emissions and economy of torch ignition stratified-charge engines (Honda CVCC) recorded by Honda.

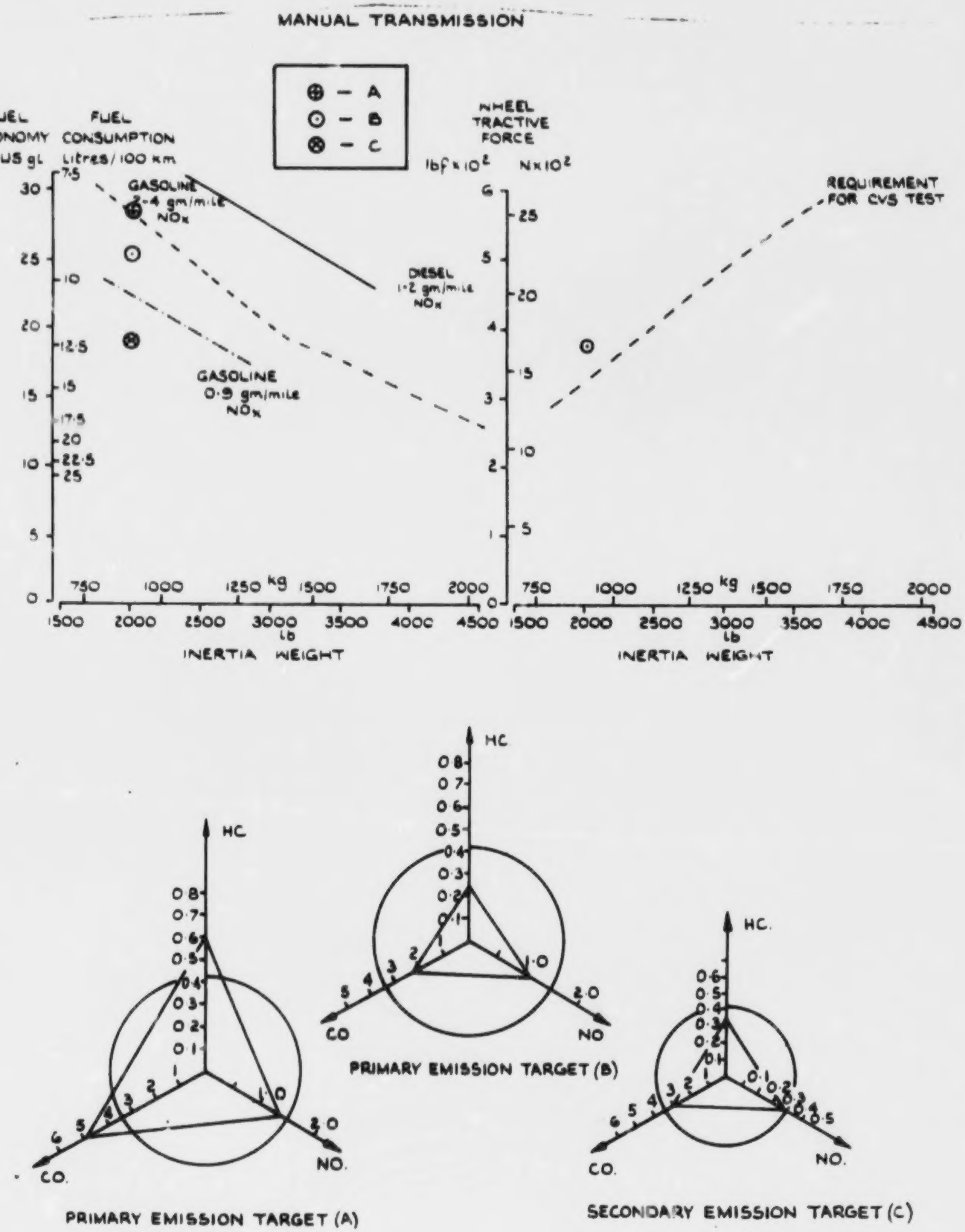
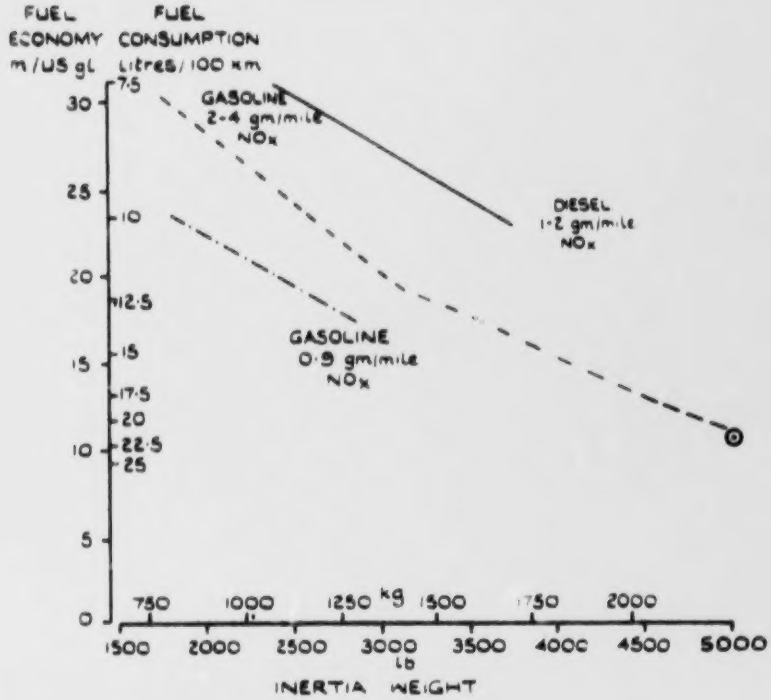


Figure 5.18. Emissions and economy of Ford scavenged prechamber engine with inboard reactor at 30,000 miles.



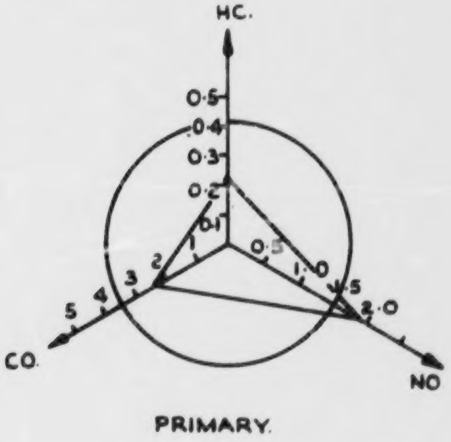


Figure 5.19. Emissions and economy of Honda CVCC car with components removed (recorded by Ricardo).

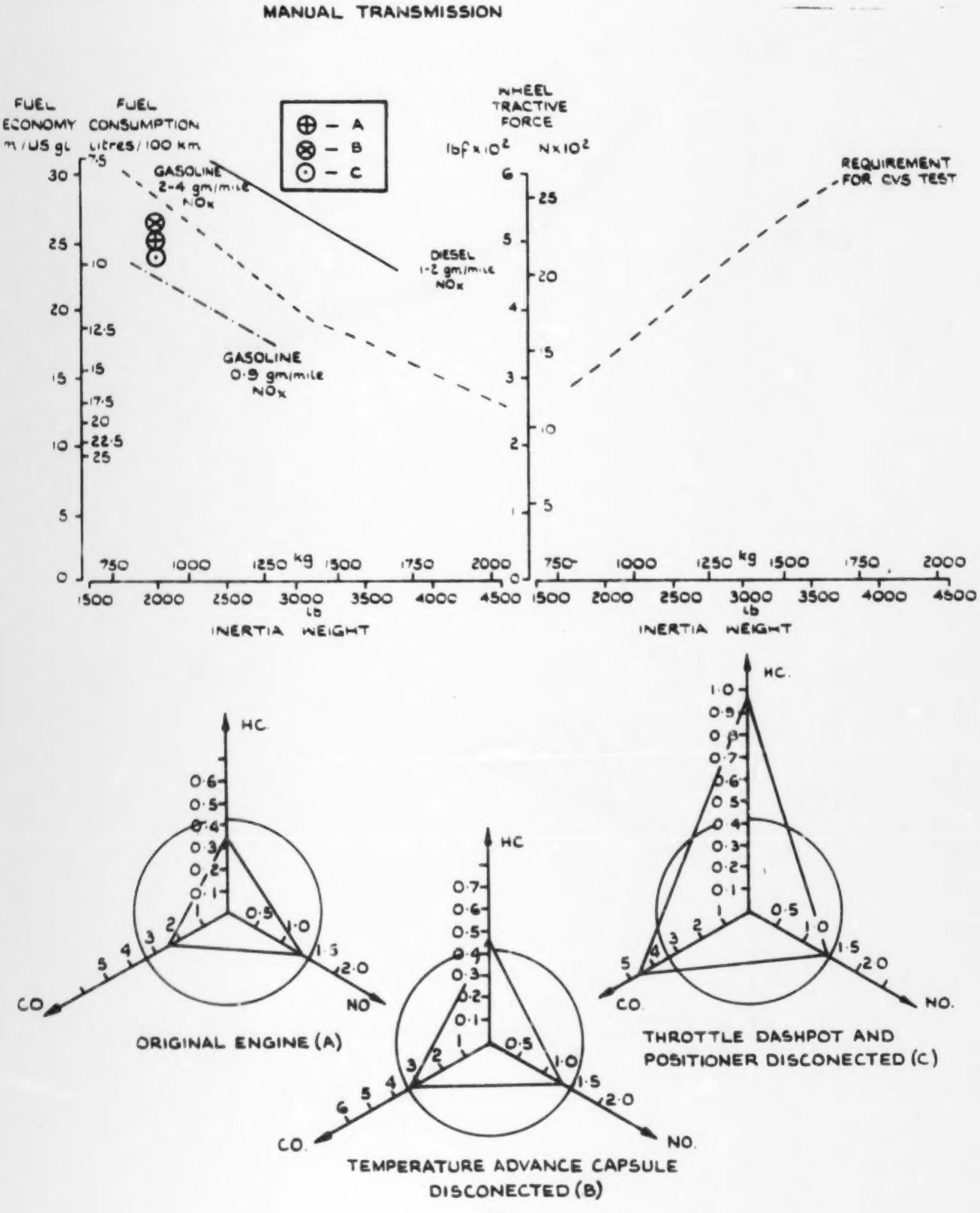


Figure 5.20. Relationship between HC and NO_x emissions and fuel economy.

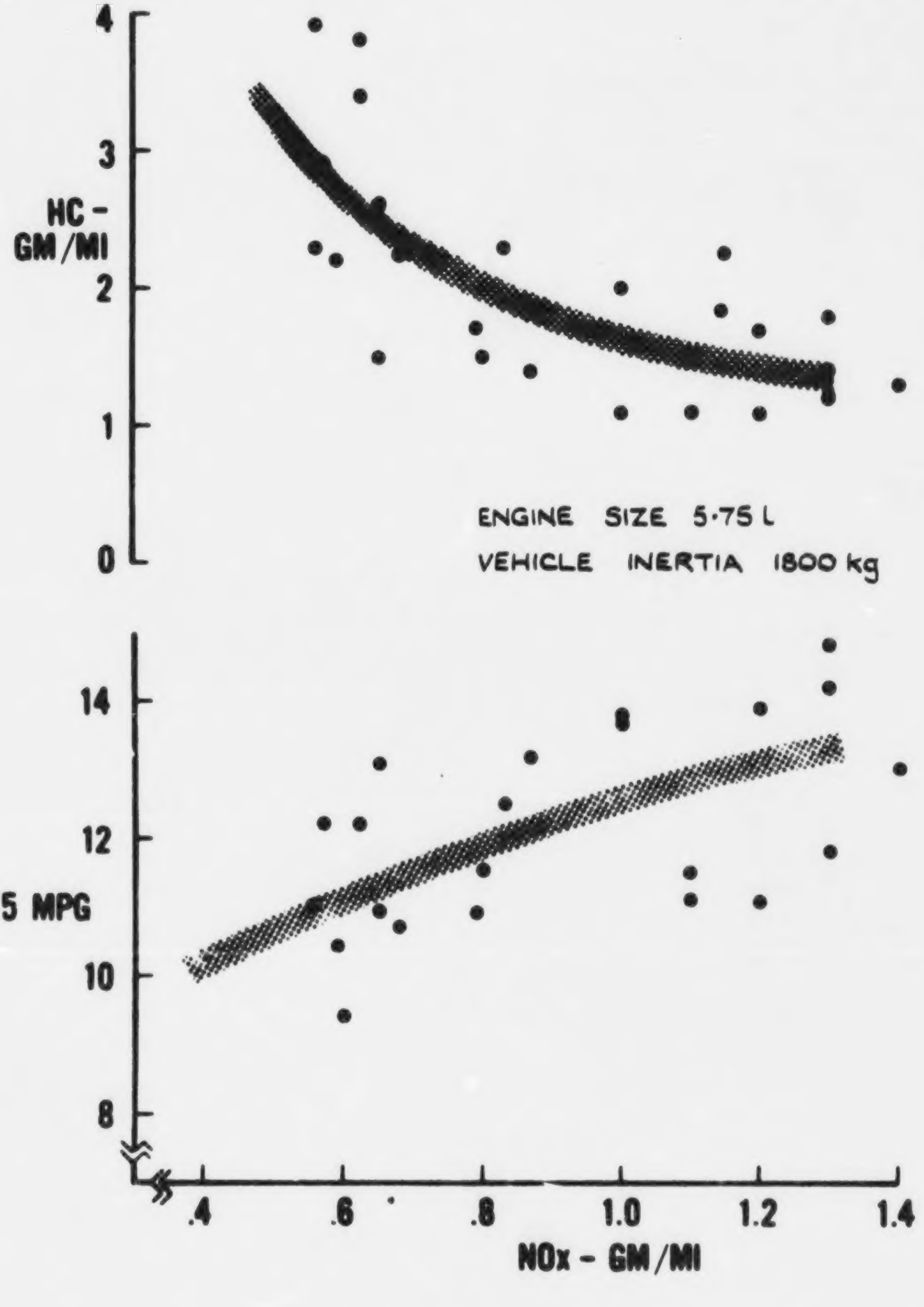


Figure 5.21. Comparison of particulate pollutants from vehicles fitted with production and prototype emission control devices.

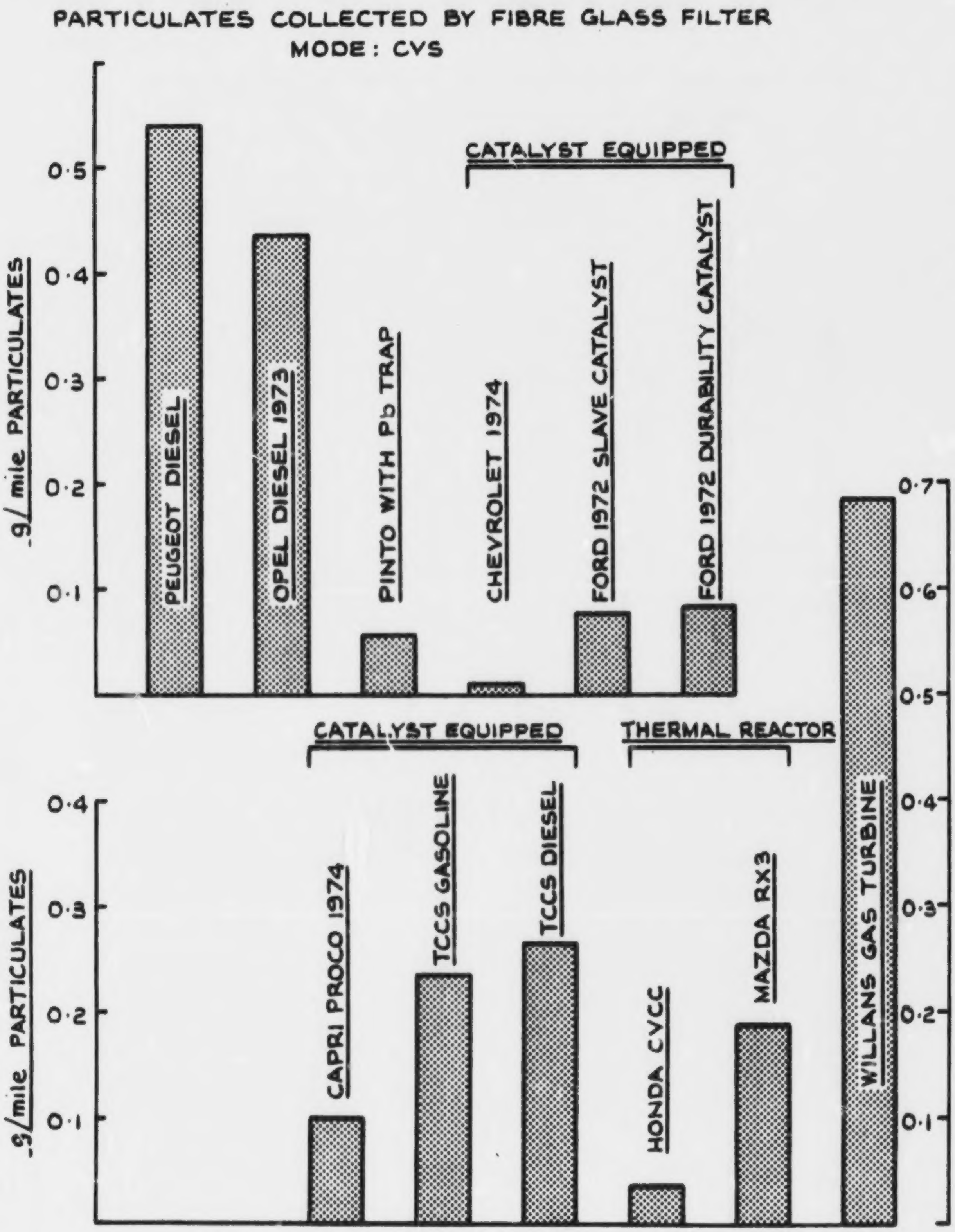


Figure 5.22. Comparison of particulate pollutants from vehicles fitted with production and prototype emission control devices.

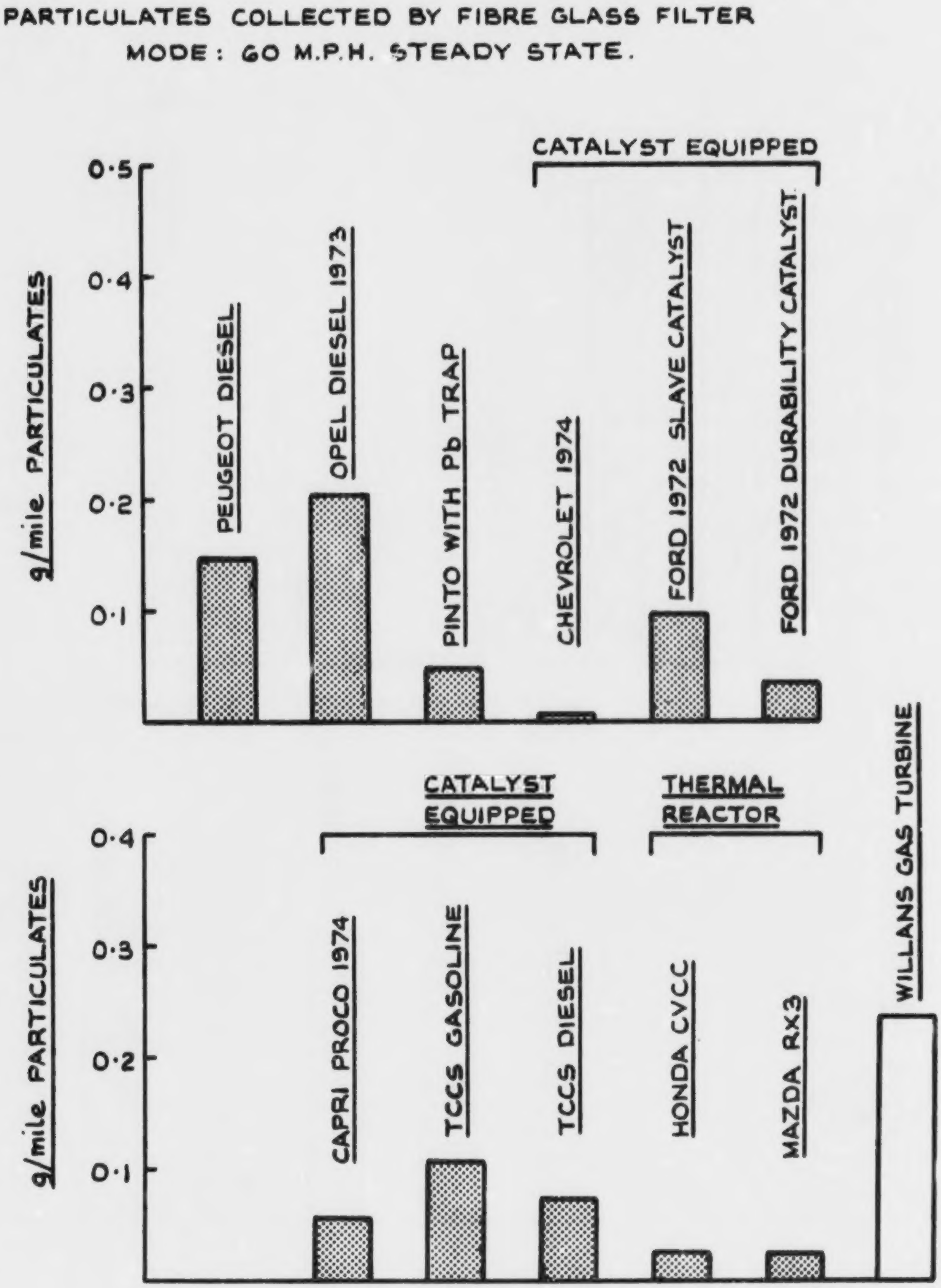


Figure 5.23. Comparison of aldehyde emissions from vehicles fitted with production and prototype emission control devices.

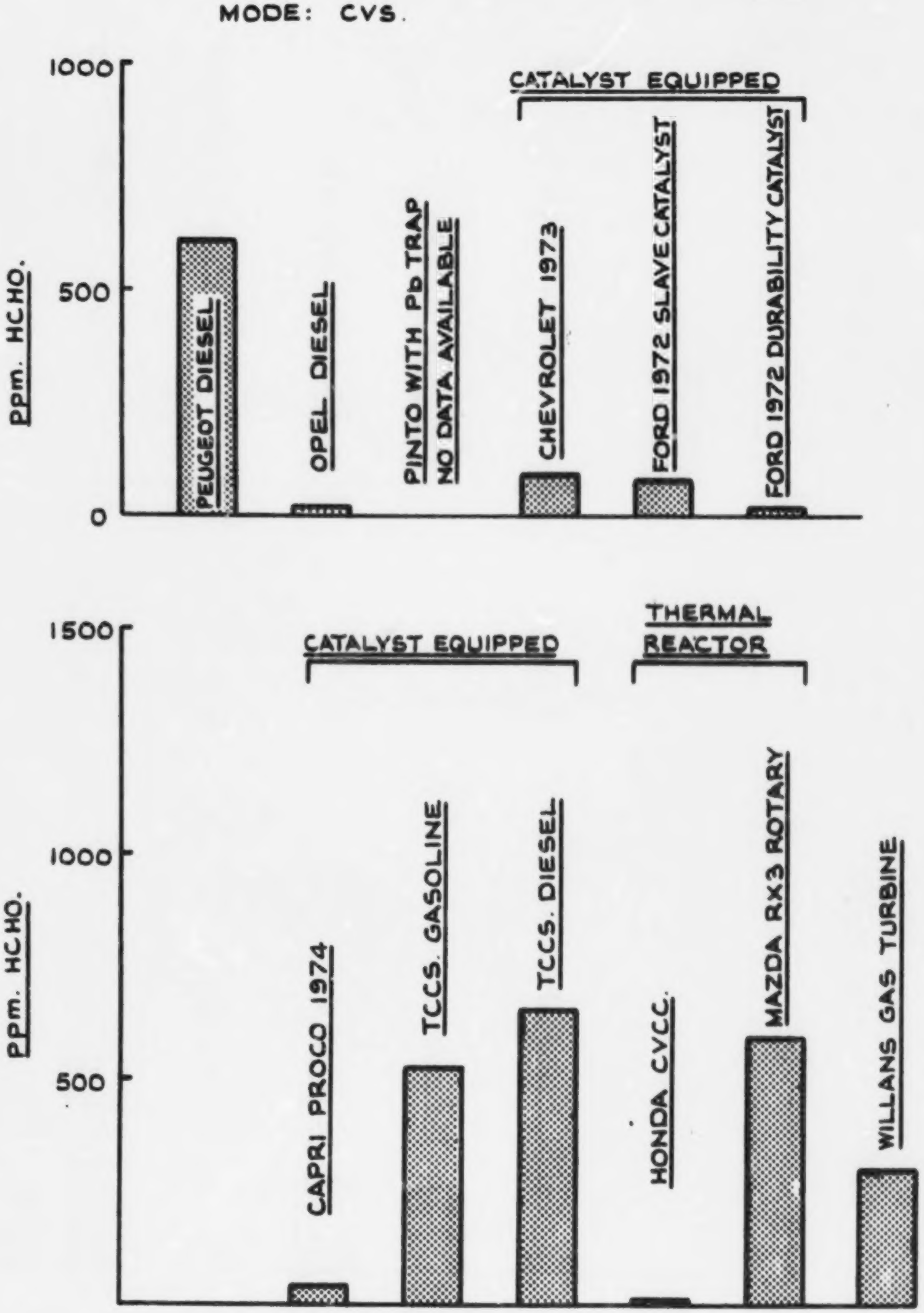


Figure 5.24. Comparison of aldehyde emissions from vehicles fitted with production and prototype emission control devices.

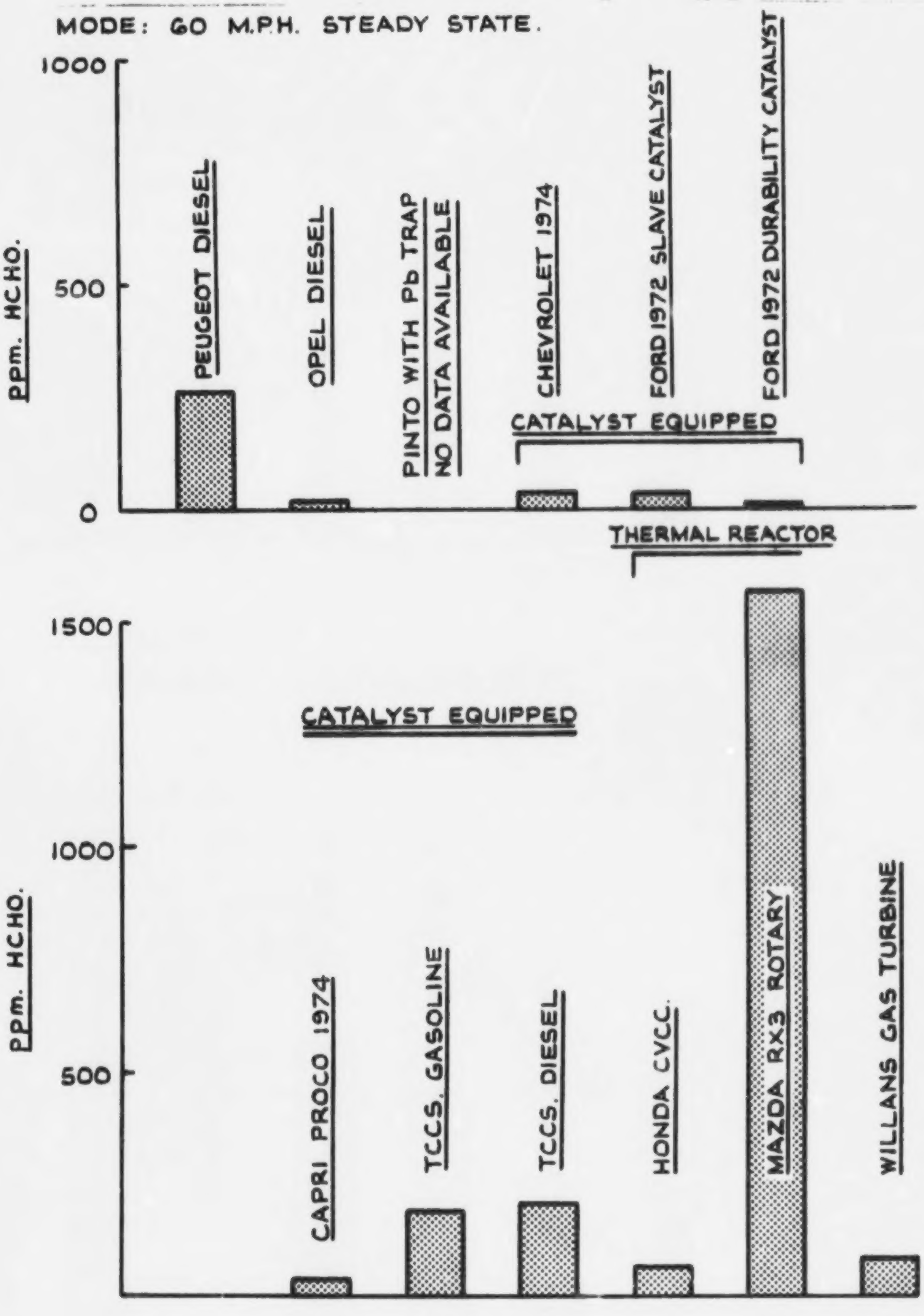


Figure 5.25. Comparison of NH_3 emissions from vehicles fitted with production and prototype emission control devices.

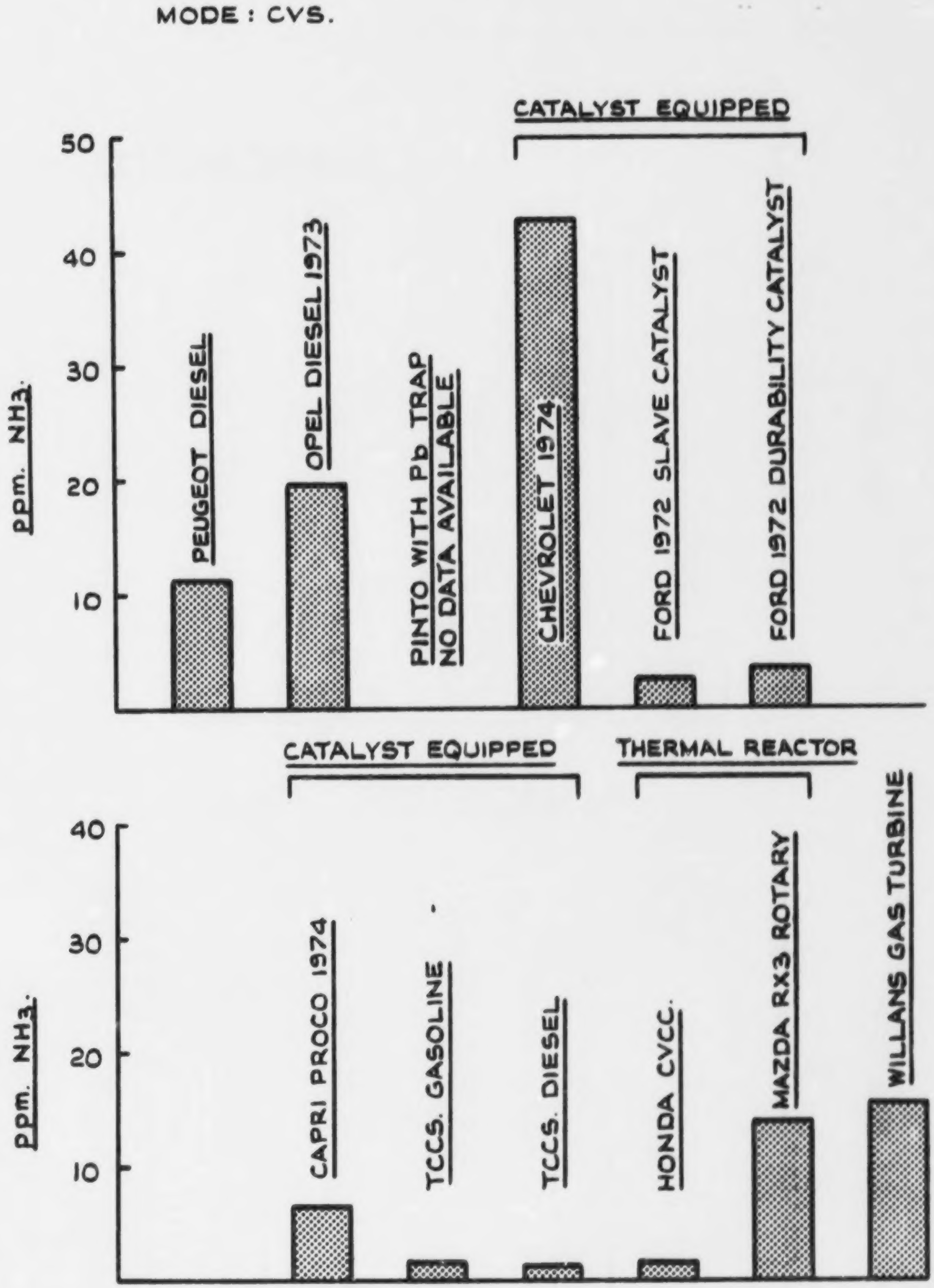
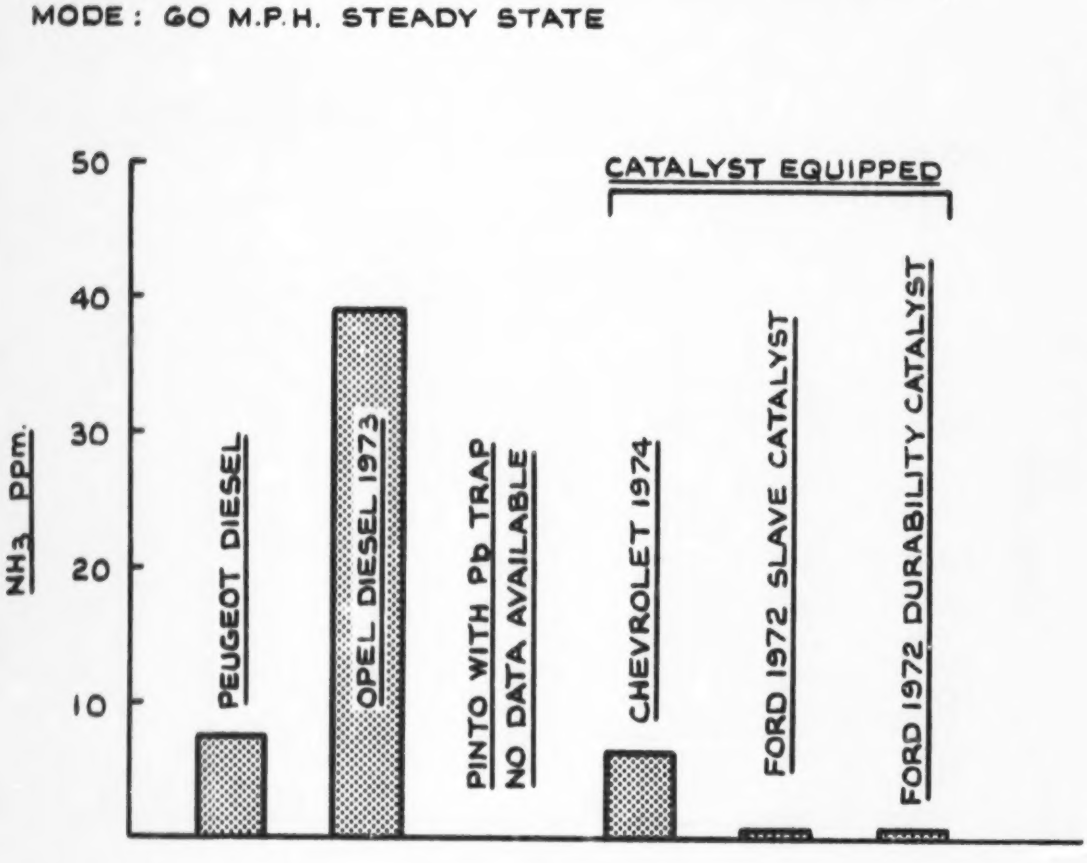


Figure 5.26. Comparison of NH_3 emissions from vehicles fitted with production and prototype emission control devices.



CATEGORY VI

SINGLE COMBUSTION CHAMBER WITH CARBURETTED MIXTURE

History

The only example of this category of system is the I.F.P. (Institut Francais du Petrole) process developed during the early 1960's, the most recent data having been published in 1965.

Topic Area (a) Combustion Chamber and Stratification Technique

The system was developed as a means of operating at lean mixture strengths without using a pre-chamber system as this results in extra heat transfer losses. The system produces a heterogeneous mixture within the combustion chamber without resorting to fuel injection or air swirl. This is achieved by dividing the carburetted mixture into two separate streams of widely differing air-fuel ratios, see Fig. 6.1. However when full power is required the engine can be supplied with a homogeneous charge so that the power output would be comparable with a conventional spark ignition engine.

A small diameter tube within the normal intake port was used to discharge a rich mixture in the vicinity of the spark plug. A lean mixture or pure air was provided by the normal inlet port. An alternative method of feeding the rich mixture into the combustion chamber is shown in Fig. 6.2, incorporating a hollow inlet valve. The flow of mixture was timed by the alignment of ports in the valve guide and valve stem and the valve was located to avoid rotation. The valve stem diameter was increased to allow sufficient flow area and the mixture was discharged from the side of the valve directed at the spark plug. The use of timed mixture flow was found necessary on the multi-cylinder engine as it was not possible to maintain heterogenity at low intake manifold pressures.

The original development of the system was carried out on a single cylinder IFP-Renault CNRS, 82.5 mm x 91.5 mm, 0.49 litre (30 in^3) engine, featuring a hemispherical combustion chamber, ref. 6.1.2. The system has also been applied to a two cylinder 85.6 mm bore and four cylinder 78 mm bore hemispherical combustion chamber engines and some exploratory tests were carried out on a non-hemispherical chamber. These latter results indicated more difficulty in obtaining the degree of heterogeneity in this kind of chamber.

Carburation

The method by which the rich mixture was fed into the chamber was an important factor in this system. The original single cylinder engine tests used a rich mixture tube fed from a separate carburettor. The orientation of the tube exit, its shape and distance from the valve head were all found to be critical factors. The presence of the tube in the inlet port had an effect on the engine when it was operating at homogeneous mixture strengths at full throttle. This was due to turbulence created by the tube and a slight lowering of the volumetric efficiency due to the restricting of the port. Although the tube diameter was only 6 mm, no problems were experienced with fuel deposits during the testing.

The application of the system to a 4 cylinder, 78 mm x 100 mm, 1.91 litre (116.5 in^3) engine revealed the problem of obtaining a controlled flow from the rich mixture tube under the low pressure conditions achieved in a multi-cylinder engine manifold. To overcome this, timed operation of the rich mixture flow was required so that it could not flow into the manifold during the inlet valve closed period, ref. 6.1.1. This system survived prolonged engine testing and no effect on volumetric efficiency was noted.

Although the published results from the single cylinder engine were all carried out with two manually controlled carburettors, a single carburettor system was described in an IFP patent. One butterfly valve controlled the flow of the rich mixture and another the main air charge during the part load throttled operation of the engine. During unthrottled operation a third butterfly valve was used which diverted a portion of the rich mixture into the main pipe.

At the time of the last published information on the IFP system it was claimed that the problems of transient response with the carburation system had been overcome and that under accelerating conditions the response of the engine was equivalent to a homogeneous charge unit, ref. 6.1.1. However, no transient test data or similar information is available to show whether these claims are justified and whether the heterogeneous mixture could be maintained during non-steady state conditions.

Ignition

The position of the spark plug relative to the rich mixture tube is important and tests on the IFP single cylinder engine showed this effect by using four different positions for the plug. This showed that there was a maximum effect in one position only and that the heterogeneity diminished as the distance from the mixture tube increased ref. 6.1.2. The ignition system itself was standard with a conventional spark plug but due to the rich mixture in the vicinity of the plug it was able to ignite overall mixtures leaner than would normally be possible. The continuous operation of the spark plug in an area of rich mixture did not result in fouling of the electrodes.

Combustion Chamber Design

The combustion chamber used on the engines on which the majority of tests were conducted was of a hemispherical form. This suited the system well as it was relatively easy to direct the rich mixture at the spark plug and the lack of squish did not cause much mixing of the charge during compression. Whether the system can be applied so easily to other forms of combustion chamber is not know, but it does seem likely that any turbulence within the chamber will reduce the heterogeneity of the mixture. The compression ratio used on the single cylinder engine was 7:1 and on the 4 cylinder unit 8.3:1. Comparative tests, on the single cylinder engine at 2000 rpm using 82 RON gasoline, between heterogeneous and homogeneous mixtures showed that with the former a knock limited compression ratio of 8:1 could be used whereas with the latter 7.4:1 was the maximum ref. 6.1.2. The overall mixture strength for both tests was 13:1. However the degree of optimum advance required when using heterogeneous mixtures increased rapidly at light loads where a lean mixture without throttling was used. The advance required was up to 14° crank angle greater and indicated the reduced speed of combustion with the heterogeneous and leaner mixtures. During unthrottled operation at lean mixtures, 24:1 air-fuel ratio, the rich mixture was heated to 88°C to achieve stable operation.

Throttling

At engine loads above 50% the engine operated completely unthrottled, with load changes being achieved by mixture strength variation. Although the main air supply was unthrottled some of the mixture from the rich supply was diverted to alter the degree of levels of stratification. This was found to be advantageous from fuel consumption and emission aspects. Below 50% load the engine was throttled to limit the minimum air fuel ratios to those at which stable operation could be achieved. The overall air/fuel ratio between 0 and 50% load was maintained at 21-24:1 and throttling was used to control the output. The minimum overall air fuel ratio at which it was possible to operate the single cylinder engine was 29:1. Tests on a twin cylinder hemispherical headed engine, 4 cylinder hemispherical engine and a non-hemispherical chamber engine gave minimum acceptable air fuel ratios of 25:1, 27:1 and 21:1 respectively. These air fuel ratios were significantly leaner than could normally be used with homogeneous carburation and indicated the advantage of the IFP system.

Topic Area (b) Cylinder Pressure Diagrams

The cylinder pressure diagrams published did not give details of the maximum pressures, but indicated that the diagram shape was the same as obtained with a conventional homogeneous charge system, ref.6.1.1. However the degree of cyclic irregularity at lean mixture strengths was significantly improved by the use of a heterogeneous mixture. This resulted in more regular combustion than would be obtained from a homogeneous charge system at the equivalent mixture strength.

Topic Area (c) Performance

Because the engine operates as a conventional spark ignition engine at full throttle the maximum power output was similar. The system used on the 4 cylinder engine to supply the rich mixture had no measurable effect on the maximum power output. The valve timing, volumetric efficiency and engine friction would all be similar to a conventional engine and therefore the overall performance of an IFP modified engine would be expected to be comparative with a conventional spark ignition engine.

Topic Area (d) Fuel Consumption

Data available on the fuel consumption of the IFP engine is limited to results from the single cylinder engine. No figures were available for the multi-cylinder unit under test bed or vehicle conditions. It would be expected that the fuel consumption of the IFP engine should be very similar to a conventional type at maximum power as the mixture reverts to a homogeneous charge. However as the load reduces the benefit from the heterogeneous charge should increase, assisted in part by the ability to operate unthrottled over a portion of the load range.

The single cylinder engine results indicated a gain in fuel consumption of about 5% compared with the optimum values obtained with homogeneous carburettion. As the carburation settings normally used on a conventional engine are a compromise rather than optimum the actual gain was nearer 10%. Whether this advantage could be maintained under road conditions when the transient operation of the system would have an effect on the overall fuel consumption is not known and depends largely on whether the carburation system used was capable of controlling the mixture under all conditions.

Brake fuel consumption figures scaled from the single cylinder engine are shown in fig. 6.3 and show a value similar to that of the diesel engine at medium loads. These values were not adjusted for the higher friction levels of the single cylinder engine. Although the figures for the IFP engine do not extend to very low loads it is better than the economy of the diesel engine at these conditions as well. If this trend was repeated at low loads at all speeds it would be expected that a significant gain in road fuel consumption would be obtained as this is the regime in which the engine is operated the majority of the time when used on the road.

Topic Area (e) Emissions

At the time of the development of the IFP system exhaust emission control was in its early stages. Therefore no C.V.S. emission test results were obtained and the majority of the emission data published was from the single cylinder engine. This showed that at 3000 rpm, full load, part load and no load the use of heterogeneous carburation extended the range of mixture strength over which the HC and CO levels could be maintained below a target level ref. 6.1.1. The target levels chosen were 300 ppm HC and 0.3% CO and the results are shown in fig. 6.4. The improvement achieved was further extended by the use of variable heterogeneity, the addition of some of the fuel in the main charge rather than all through the rich mixture system. The ratio of rich mixture to lean mixture could be altered for each condition but results showed that this ratio was not critical from the point of view of unburnt HC emissions. The levels of emissions produced with the IFP system were not significantly lower than with homogeneous carburation but were maintained over a much wider range of air fuel ratios. The range of mixture strength over which the normal carburetted engine will operate reduces with the degree of throttling and shifts towards stoichiometric.
This limiting of the mixture strength range with increased throttling could be partly attributed to dilution of the fresh charge by exhaust residuals. The use of variable heterogeneity did not extend the lean mixture limit but only the rich limit. Tests had indicated that when fuel was supplied by the rich mixture tube alone a condition could occur where the mixture around the spark plug was too rich which resulted in an increased fuel consumption, and would also increase HC levels in the exhaust. No values were published for NO_x levels from the system, but it is likely that these would be lower than from the conventional engine as the overall mixture used would reduce NO_x formation. The reduced exhaust temperatures might also lead to difficulties with catalyst warm-up rates if such a system was used. To increase the exhaust temperatures it would be necessary to resort to a greater degree of throttling.

Topic Area (f) Multi-fuel Capability

As the combustion system is very similar to a conventional homogeneous charge type it would not have a multi-fuel capability. It still has an octane requirement, as shown in Topic Area (a) which is only slightly different from a homogeneous charge engine.

Topic Area (g) Engine Components and Manufacturability

To apply the IFP system to an engine would require a new carburettor and modifications to the inlet manifolding, and from the reported work on the multi-cylinder engine a form of timed rich mixture system. The carburettor would be required to achieve the various degrees of heterogeneity required and the inlet manifolding would have to be designed to avoid the generation of any swirl. The system requires a quiescent chamber and any organised air motion is likely to upset the stratification. The inlet ports on some engines which induce air motion would be unsuitable. The combustion chamber requirements could also limit the types of engine to which this system could be applied. The method used to achieve the timed induction of the rich mixture charge involved alterations to the inlet valve and guide and it is possible that other methods of achieving the same effect could be designed. However any method other than a simple pipe in the inlet port will involve changes to the inlet system components. The reliability of the inlet valve controlled timed charge was claimed to be good and no problems were experienced with it. The exhaust valve life was not affected by the use of lean mixture as the ability to operate unthrottled resulted in lower exhaust temperatures than experienced with homogeneous carburation. Spark plug life was not affected by the use of the rich mixtures and no problems were experienced with fuel deposits in the rich mixture feed system. The system does not seem to present any serious problems provided the combustion chamber arrangement and spark plug position are suitable. However lack of knowledge of the detail and durability of any engine conversions make it difficult to estimate the true potential of the system, but the engine size and weight should not be increased greatly, if at all. The cost of the special carburettor and rich mixture feed system are likely to be greater than on the conventional engine but apart from this no difference would be expected.

Topic Area (k) Noise & Vibration

No measurements of engine noise have been published for the IFP system, but the noise levels should be the same as the conventional engine on which the conversion is based. The cylinder pressure diagrams indicate the normal shape expected from homogeneous charge engines and therefore no extra combustion induced noise would be expected. The degree of cyclic irregularity was significantly reduced at lean mixture strengths by an increasing degree of heterogeneity, 10% being sufficient to eliminate cycles on which combustion failed to occur at an overall mixture strength of 20:1. This reduction in cyclic irregularity should reduce engine vibration and generally make the engine more driveable in a vehicle as well as having a beneficial effect on efficiency.

Topic Area (l) Cold Starting

No details have been published regarding the cold starting ability of the IFP engines either in single or multi-cylinder form. However there does not seem to be any reason as to why it should be inferior to the conventional homogeneous spark ignited engine and it could possibly be better. This improvement could be caused by the use of the rich mixture supply system which would ensure that a rich mixture was quickly directed to the vicinity of the spark plug, rather than having to enrichen the complete charge as in the case of the conventional manifold arrangement. This could possibly dispense with the choke system normally used with carburettors.

Application to Motorcycles

The lean mixture limit of these engines would certainly be extended by using the IFP principle, although the requirements for low in cylinder gas velocities may be affected by the high engine speeds, and the stratification would be destroyed. As in Category V, it is doubtful whether the increased engine complexity of this type of stratified charge engine is justified, since the same economy and emission advantages could be obtained on the conventional gasoline engine motorcycle, by simple alterations of the engine operating parameters.

Conclusions from Category VI

1. The IFP system was the only example identified of a Category VI stratified charge engine.

2. The system could be applied fairly easily to an existing production engine, provided it had a quiescent combustion chamber, which probably limits it to

the hemispherical type.

3. Throttling was required below 50% load as there was a minimum air-fuel ratio at which stable operation could be achieved. Above this load unthrottled operation was possible with a varying percentage of the rich mixture being diverted into the unthrottled main air charge.

4. The maximum output of the engine was similar to a conventional homogeneous charge engine as the IFP system reverted to this type of operation at full load.

5. An improvement in fuel consumption especially at low loads compared with the conventional diesel engine was indicated by results from a single cylinder engine, but no road fuel consumption data was available to confirm this.

6. Exhaust emissions measured at steady states on the test bed indicated that although the minimum levels of HC and CO were no lower than from the same engine with conventional carburation, the use of the IFP system enabled the minimum levels to be achieved over a wider spread of mixture strengths. No data was available for NO_x emission levels although this is likely to be low, and the lack of any C.V.S. emission data made it impossible to evaluate the performance of the system under transient conditions.

7. As the engine operated under some conditions as a conventional homogeneous charge engine it did not have a multi-fuel capability.

8. The system could be applied to an engine without the necessity for extensive structural modifications, but a special carburettor, mixture control system and special inlet valve were required, all of which needed development. The durability of these components was unlikely to be a problem, but no road durability data was available from the published references.

Figure 6.1 I.F.P. engine.

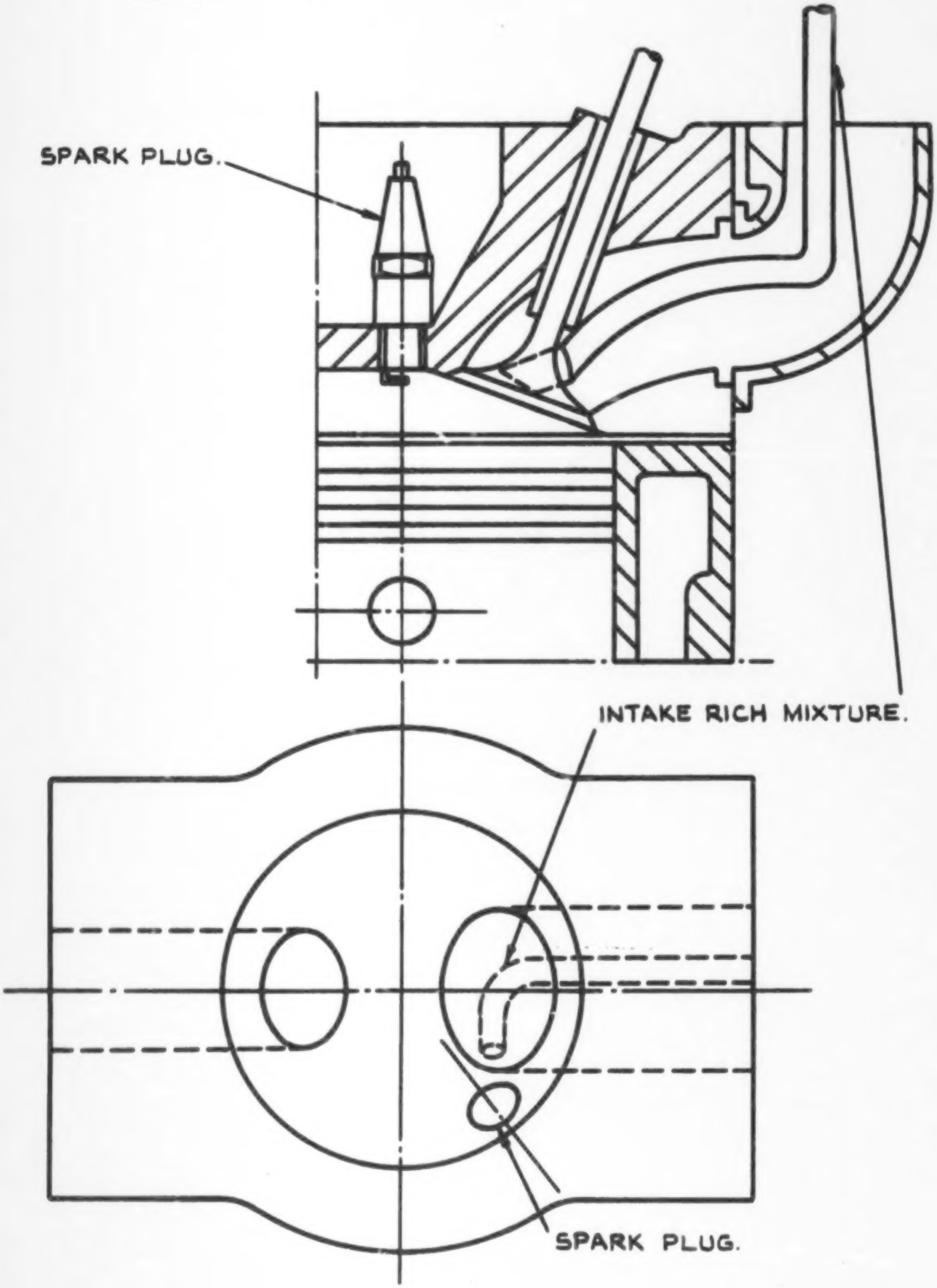

SPARK PLUG.
INTAKE RICH MIXTURE.
SPARK PLUG.

Figure 6.2 I.F.P. applied to four cylinder engine.

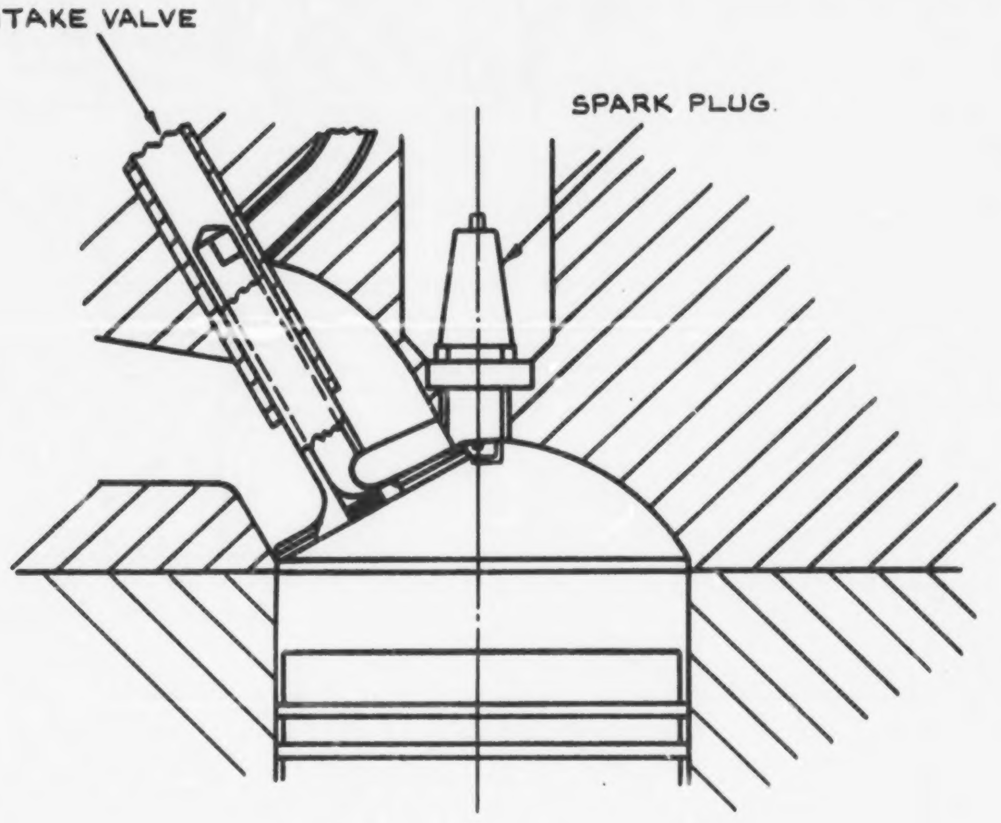

INTAKE VALVE
SPARK PLUG

Figure 6.3. Comparison of fuel consumption of diesel engine and I.F.P. engine extrapolated to target horse power.

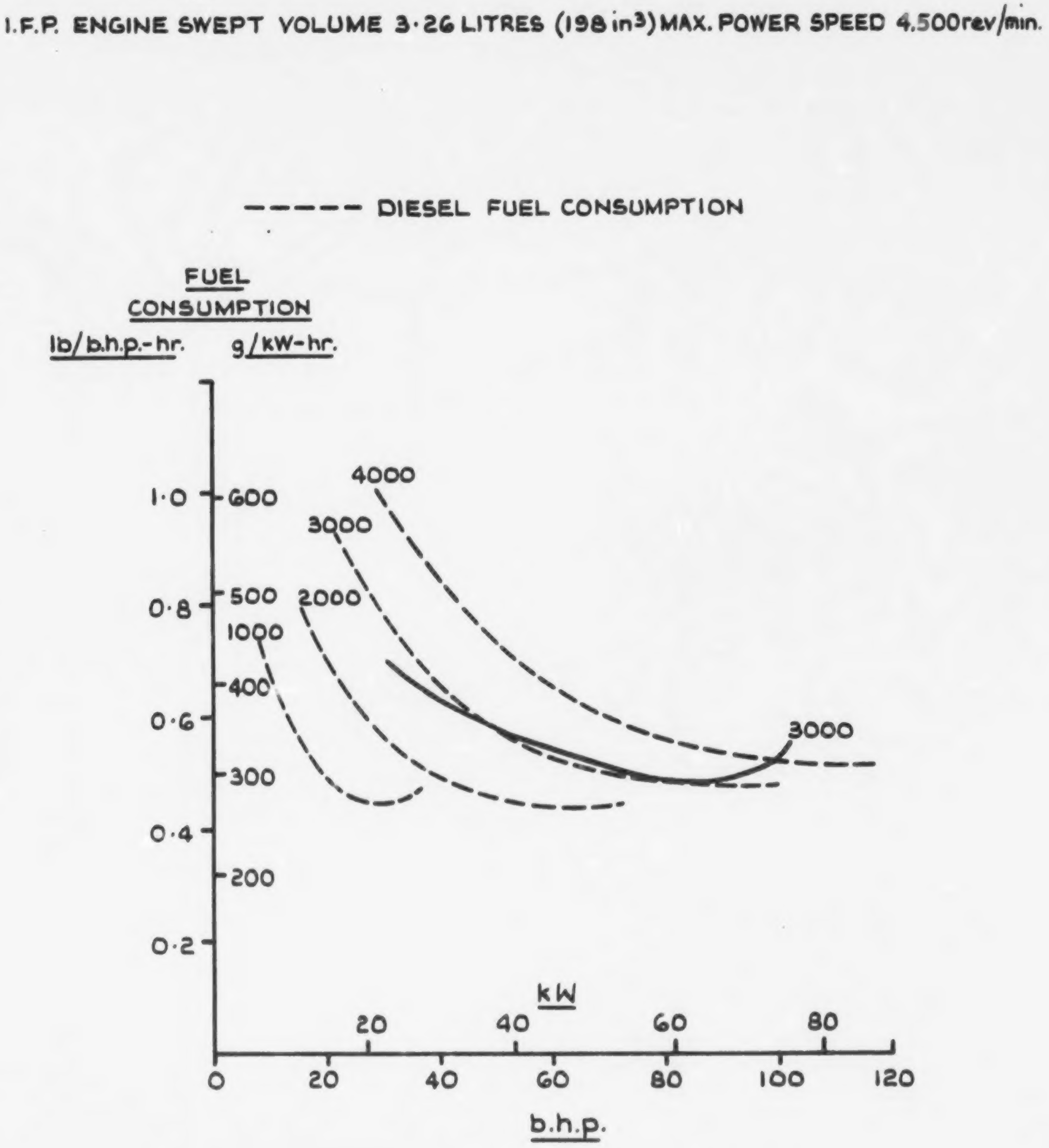
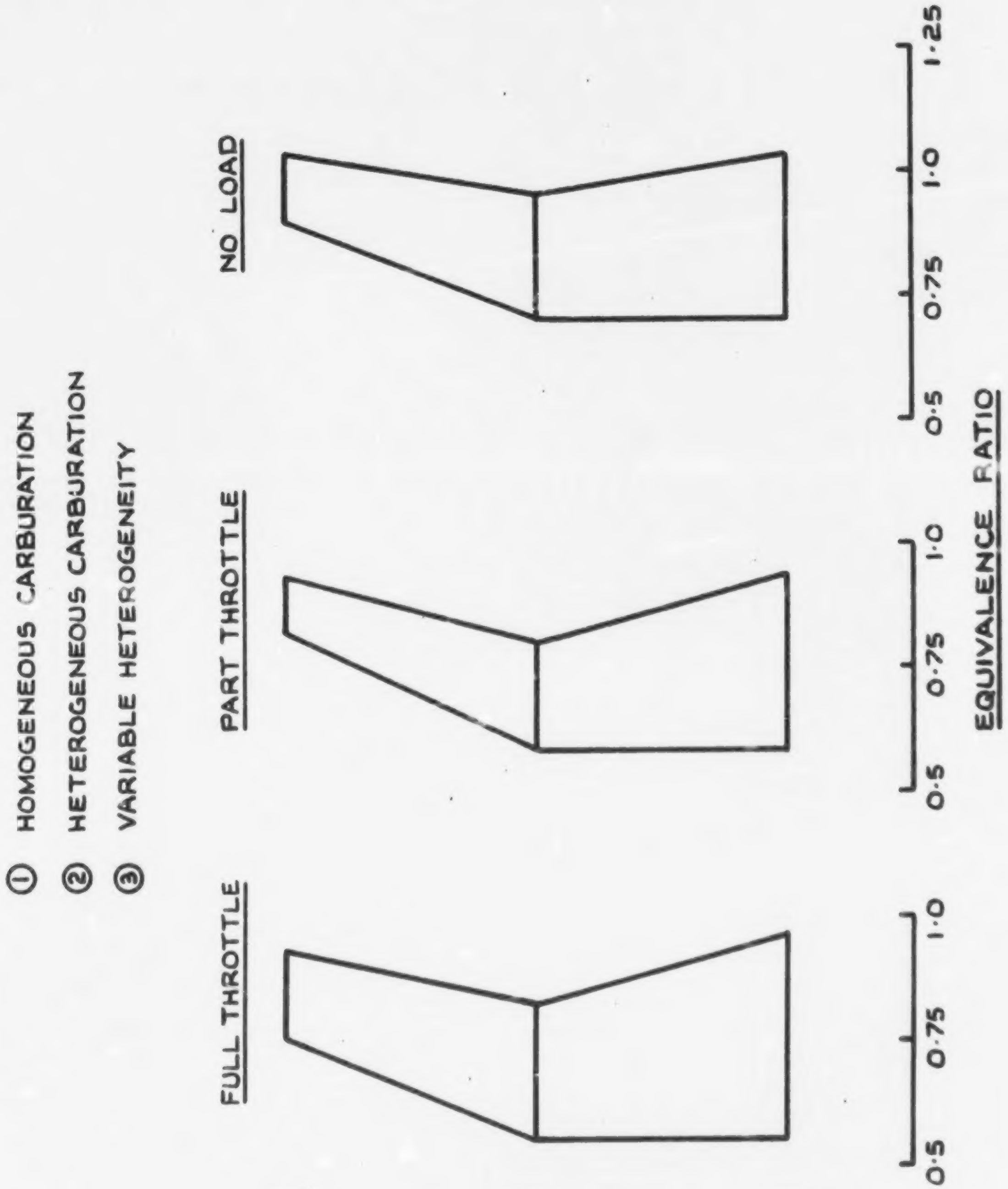


Figure 6.4. I.F.P.-Renault CNRS engine.

ZONES OF EQUIVALENCE RATIO LIMITED BY 300 ppm. HC
AND 0·3% CO. FOR THREE TYPES OF MIXTURE PREPARATION
ENGINE SPEED 3000 rev/min.



CATEGORY VII

TWO STROKES, EXHAUST DILUENT AND MISCELLANEOUS ENGINES

History

The systems reviewed in this section can be sub-divided as follows:-

two strokes;	NICE, YOCP, Kataoka & Hirako, Clawson, Heintz, Fairbanks Morse and Ricardo.
exhaust diluent;	Jessel
miscellaneous;	Kushul, Cranfield-Kushul and Ricardo.

These systems have been applied to a wide range of engines having bore sizes from 60 mm to 206 mm, piston controlled ports, poppet valve, sleeve valve, two stroke and four stroke cycle. The only system currently in production is the Fairbanks Morse which is used on a stationary gas engine operating at speeds of 500 to 900 rpm. The Kushul engine was fully evaluated on the test bed and was operated in a vehicle before 1965, but details of the more recent history of this engine are not available. The NICE and Clawson engines have been tested in vehicles but apart from these the other systems considered have never progressed beyond the test bed development stage. The systems described are illustrated in fig. 7.1-7.6

Topic Area (a) Combustion Chamber and Stratification Technique

Due to the wide range of engines types considered each system will be described in detail.

The Nippon Clean Engine - NICE relies on air motion induced by three transfer ports, a wedge shape combustion chamber and an exhaust control valve to achieve charge stratification ref. 7.1.6. The mixture is supplied by a carburettor and fed into the crankcase via a reed valve. The third transfer port was positioned to give the required air motion as a result of rig tests using a glass cylinder and head. The wedge shape combustion chamber is an important part of the system and the actual dimensions of the chamber and squish area require optimisation for each cylinder size. The transfer passages are designed to maintain a constant mixture velocity flowing into the cylinder unlike the normal 2 stroke arrangement where the passage is slightly throttled to increase the mixture velocity. This constant velocity mixture flow reduces the amount of dispersion and dilution with the residual gases to a minimum. The exhaust control valve reduces turbulence in the cylinder as a result of rapid exhaust gas blow down and prevents the gas from diffusing. This valve in conjunction with the velocity controlled transfer ports and the combustion chamber shape combine to form the stratification pattern. The control of the charge motion within the cylinder appears to be the important controlling factor in this system and the addition of the fuel to the air does not have a significant effect. However the intake and exhaust are throttled across the speed range including idle, to achieve this effect. (fig. 7.1).

The Yui and Ohnishi Combustion Process - YOCP engine, utilises the air motion induced in the cylinder by the Schnurle system of piston controlled ports to achieve stratification ref. 7.1.3. Direct fuel injection into a wedge shape combustion chamber was used, at timings of between 130 and 80° BTDC at 2000 rpm, dependent on load. The four injector sprays were in the opposite direction to the air motion and this resulted in the fuel centrifuging to the outer layer surrounding a region of leaner mixture and exhaust residuals. This air motion was at right angles to the vertical axis of the cylinder and the shape of the combustion chamber directs the rich outer layer towards the twin spark plugs aided by a 53% squish area. Although a small amount of fuel was deposited on the chamber walls this quickly evaporated but if the injector was positioned so that the fuel sprays impinged directly on the chamber wall the level of unburnt hydrocarbons in the exhaust increased indicating that the method of combustion is different from the MAN system of evaporating the fuel off the hot piston surface. The engine is throttled during light load operation only. (fig. 7.2).

The Fairbanks Morse system, the only one in this section at present in production, is applied to a stationery gas engine ref. G23. This is a four cylinder opposed piston two stroke type operating at speeds between 500 and 900 rpm. Two small pre-chambers are utilised per cylinder and these are each of 1.5% of the total clearance volume. The pre-chamber does not have a separate inlet valve and is scavenged by the main cylinder. The spark plug is positioned in the pre-chamber as is a gas injector which provides a near stoichiometric mixture in the vicinity of the spark plug. The majority of the fuel is admitted into the cylinder through the gas valve and is ignited by torch ignition from the pre-chambers. This enables ignition of the main chamber charge to be achieved at overall air fuel ratios of 30-40:1 across the load range, and alleviated the detonation problems frequently encountered in open chamber gas engines when operating at the minimum fuel consumption point. (fig. 7.3).

The Kataoka and Hirako - K. & H. system uses a cylindrical unscavenged pre-chamber applied to a loop scavenged (Schnurle) system two stroke engine, ref. 7.1.5. The pre-chamber volume is 14% of the clearance volume and is separated from the main chamber by a small orifice. Fuel is injected directly into both the pre-chamber and the main chamber, the spark plug being situated in the former. Test results have only been reported at 2000 rpm using a main chamber injection timing of 90° BTDC and a pre-chamber injection timing of 60° BTDC, with an ignition timing of 5° BTDC. At idling fuel is injected into the pre-chamber only. Above this load fuel is added to the main chamber without the use of throttling. However present results are limited to steady state conditions at 2000 rpm only. (fig.7.1).

The Clawson system; which has been applied to a 3 cylinder 2 stroke SAAB engine, utilises a spherical pre-chamber connected to the main chamber by an orifice, refs. G23 & 7.1.4. The pre-chamber volume is 15% of the clearance volume and a fuel injector and spark plug are located in it. The injector is supplied with a constant rate of fuelling and additional fuel supplied to the main chamber by an electronically controlled injector. The passage of air through the orifice into the pre-chamber creates turbulence which mixes the air, residuals and fuel into a uniform mixture. The air fuel ratio in the pre-chamber is always rich and the main chamber lean, and the operation is unthrottled at al. conditions.

The Heintz Ram Straticharge system; has been applied to a 1.78 litre (108.6 in^3) V4 and a 0.95 litre (58 in^3) 3 cyl. two stroke engine, ref. G23. It incorporates a pre-chamber with a separate inlet valve which is supplied with air from the second stage of a two stage Roots blower. The main charge is supplied through the conventional ports from the primary stage of the blower. The fuel is injected continuously into the pre-chamber air supply which is throttled at light load operation. When the pre-chamber valve opens, at about BDC a rich mixture is induced. At the same time air only is being induced through the intake ports. Some of the rich mixture spills into the main chamber to form a very lean mixture and as the piston moves up the bore part of the main chamber mixture is forced back into the pre-chamber. The high degree of swirl caused by the chamber geometry forces fuel to the periphery of the pre-chamber forming an ignitable mixture in the vicinity of the spark plug. (fig. 7.2).

The Ricardo E65 engine of the late 1930's was a sleeve valve two stroke engine of 1.67 litre (102 in^3)/cylinder, ref. G25. The fuel was directly injected into a bulb shaped chamber in the cylinder head, the volume of this bulb being approximately 20% of the clearance volume. The main fuel injection occurred as the inlet ports were uncovered and gave thorough mixing of the air and fuel. The secondary injection occurred at the end of the main injection and was a coarse low pressure wide angle spray which remained within the bulb and provided a rich mixture in the vicinity of the twin spark plugs positioned in it. At idle the secondary injection alone was sufficient and the intake was unthrottled. At full power the primary and secondary injection gave a mixture strength rich of stoichiometric, but at reduced loads the mixture strength was leaner than that which could normally have been ignited. Up to 15% and above 40% full torque the engine operated in a stable manner, but between 15 and 40% when the mixture in the main chamber was too lean to ignite regularly,the combustion was erratic. In aircraft operation for which the engine was intended this was not a serious problem as it was not required to operate in the 15 to 40% range, but this problem was not overcome, although throttling was not applied.

The Ricardo APC-1 was also a sleeve valve two stroke unit but the cylinder size of 0.34 litre (21 in^3) was considerably smaller than the E65 ref. 6.15. This engine had a similar combustion system to the E65 and was developed during the late 1940's. (fig. 7.3)

The only example of the exhaust diluent engine studied was that developed by Jessel, Uyehara and Myers, based on U.S. Patent No. 3944336 granted to the latter two gentlemen, ref. 2.2.2. This patent describes the system applied to both 2 and 4 stroke engines, but the Jessel engine was a 4 stroke application based on a CFR engine, ref. 7.2.1. The system attempts to achieve stratification by recirculating a proportion of the exhaust gas back into the cylinder and only partially filling the cylinder with combustible mixture. Therefore power can be controlled by the volume of cylinder occupied by recirculated gas and the recombustion of the gas should reduce hydrocarbon and CO emission levels. At the start of the induction stroke a certain volume of exhaust is allowed into the cylinder through a special valve. Later in the stroke the intake valve allows in a mixture of air and fuel which remain in a pocket in which the spark plug is positioned. During compression the ignitable mixture is retained in this pocket and the exhaust products remain in the main part of the chamber. Throughout combustion the two zones remain stratified and the recycled exhaust is reheated. At the start of the exhaust stroke the gas in the vicinity of the exhaust valve, which is also located in the pocket, is that which has only been combusted once. When the exhaust valve opens this gas is recirculated back to the recirculation valve and into the cylinder for recombustion. After the required volume of gas has been recycled a flapper valve in the exhaust diverts the remainder of the gas to waste. The timing of the flapper valve is intended to recirculate only the residuals resulting from the fresh charge just burnt and to allow the recycled residuals to pass to waste. (fig. 7.5).

A similar effect could be achieved by using a 6 stroke cycle and brief description of such an engine developed by the Kanao Kinsoku Kogyo Co. Ltd., in Japan has been reported, but no details have been located ref. 7.2.3.

The Kushul engine was developed in Russia in the early 1960's and features a pair of cylinders connected by a transfer port. The mixture cylinder which contains the spark plug leads the air cylinder by 20 - 30° crank angle. Air and fuel mixture are drawn into the mixture cylinder and air only into the air cylinder. The air cylinder must have a high compression but this is limited by practical considerations. When combustion starts in the mixture cylinder the air piston is still rising and it has not reached its maximum compression pressure. As the mixture piston begins to descend the pressure in the air cylinder exceeds that of the mixture cylinder and the air is transferred into the burning mixture. The geometry and dimensions of the throat are such that the air and burning gases mix vigorously and cause complete combustion. Combustion products can spread into the air piston as expansion occurs, but these are exhausted from valves in both cylinders during the exhaust stroke. Some recent work has been carried out on this type of engine in the United Kingdom and it has been called the Cranfield-Kushul engine to distinguish it, ref. 7.4.1. However this work is at a very early stage compared with the original Russian developments which culminated in 30,000 km of road operation. (fig. 7.6).

Early Ricardo work on 4 stroke sleeve valve engines led to a stratified charge scheme which was applied to the E30 2.7 litre (166 in^3) single cylinder engine in 1924, ref. 7.3.1. In addition to the main inlet ports at the top of the liner an additional row was used at the bottom of the liner which was only uncovered by the piston at the end of the induction stroke. When the engine was operating in a throttled condition and drawing mixture through the main ports a certain depression was present in the cylinder when the lower ports were uncovered. This depression varied with the amount of throttling being applied. When the low ports were open extra air was drawn in which remained above the piston crown and did not mix with the main charge, during compression. As expansion occurred the two charges mixed and became a homogeneous and cool mixture. To avoid excessive dilution the two throttles were interconnected to throttle the air supply progressively below half load. This system enabled the engine to operate over a much wider load range than was possible before the secondary ports were used. (fig. 7.4).

Fuel Injection

The NICE engine was the only two stroke system that did not use fuel injection as part of the combustion system. The degree of mixture preparation required was achieved by the use of a carburettor and reed valve feeding into the crankcase. Three other systems, YOCP, Kataoka and Clawson used direct in-cylinder injection, in most cases adapted from other engines and therefore not necessarily fully developed for the particular application.

The YOCP use a Bosch gasoline injection pump originally from a Daihatsu engine and an injector from a Nissan diesel engine. The injection pressure used was approximately 100 kg/cm^2 (1400 lb/in^2) due to the limitations of the equipment but it was considered that it should operate at a pressure of 40-60 kg/cm^2 (570-850 lb/in^2). The start of injection timing range was 130° to 80° BTDC across the load range at 2000 rpm, this being the only speed for which results have been published.

Two high pressure Bosch gasoline injection pumps were used on the Kataoka and Hirako engine the injection pressure being 44 kg/cm^2 (630 lb/in^2). The injector spray angles used were a 30° cone in the pre-chamber and a 50° cone in the main chamber. The end of injection timing was fixed at 60° BTDC and 90° BTDC for the pre and main chambers respectively at 2000 rpm, again the only condition at which results were obtained.

The Clawson system used two separate low pressure Bosch fuel injection systems, that to the pre-chamber being of fixed quantity delivery, as was the Kataoka and Hirako, and the main chamber supply being electronically controlled. No details of injection pressure and timing have been published, but the control system was developed to the required standard to allow an engine in a vehicle to be evaluated over the Federal CVS emission test cycle.

The Heintz system used a continuous low pressure injection system feeding into the intake port upstream of the pre-chamber inlet valve. The fuel flow control was achieved by varying the fuel line pressure. This system was not satisfactory as the fuel accumulated during the valve closed period entered the cylinder as a slug and passed out of the pre-chamber and main chamber into the exhaust, adversely affecting both HC and fuel consumption. This problem could possibly be alleviated by attention to detail in the fuel injection system.

In the Ricardo E65 and APC-1 engines a single direct injector was used, whereas in the Fairbanks Morse engine a single gas injector in each of the two pre-chambers per cylinder was required.

The Jessel engine was operated on ethane and no special mixture preparation was used. On the Cranfield-Kushul engine modified carburettors were used, whereas on the original engine both carburettors and direct fuel injection had been assessed. The best performance figures published were achieved with carburettors, but the fuel injection had been used to allow multi-fuel operation.

Ignition

One of the main reasons for misfire at idle and low loads in the two stroke is the lack of combustible mixture in the vicinity of the spark plug. Poor mixture distribution within the cylinder and dilution with exhaust residuals result in erratic operation. The aims of most of the two stroke systems reviewed has been to improve the idle and low load operation by ensuring that the spark plug is always surrounded by a region of ignitable mixture even if this is not typical of the rest of the cylinder. With the exception of the YOCP engine none reported the use of special ignition systems. The YOCP engine used a transistorised capacity discharge system with two coils connected in parallel giving a generated voltage of 40 kW, to fire two spark plugs simultaneously. Spark plugs used were also of conventional design with no system requiring the extended electrode type, therefore electrode erosion should not become a problem. It is unlikely that any benefit will be obtained in using high energy ignition systems on these type of engines as in all cases the mixture in the vicinity of the spark plug is intended to be approximately stoichiometric if the system is functioning correctly.

Air Swirl

In the two open chamber, piston port controlled engines the swirl generated by the incoming air was used to achieve stratification. In the NICE and YOCP engines swirl was initially generated by the transfer port angle: the NiCE engine used an extra transfer port designed as a result of flow visualisation tests whereas the YOCP relied on the conventional Schnurle port system. The air swirl was at right angles to the cylinder vertical axis at the start of the compression stroke and as squish commences this moves through 90° shortly before TDC in the NiCE engine. The shape of the cylinder head and combustion is designed to encourage this motion. The YOCP relies on the air motion being at right angles to the vertical axis during the period of injection after which the fuel droplets are directed into the vicinity of the spark plug by the piston squish and the combustion chamber shape. The injection is in the opposite direction to the air rotation. The various pre-chamber two stroke systems do not rely on air swirl for stratification, but turbulence generated by the orifices to the pre-chamber are used to promote fuel and air mixing.

In the Ricardo E30 and Jessel systems the two zones are passed into the cylinder through different inlets and the lack of swirl is used to keep them from mixing. Likewise in the Kushul engine the air and air/fuel regions are intended to remain separate and any violent air movement could disturb this situation.

Compression Ratio

The compression ratios used in the engines in this section cover a wide range. For the piston port controlled 2 stroke engines the compression ratios are quoted in two ways, the theoretical, assuming that compression starts to occur immediately after BDC and the effective, assuming compression does not occur until the exhaust ports have closed, usually about 70° ABDC. The quoted compression ratios were as follows:-

	Effective CR
NiCE	6.5:1
YOCP	10:1
K. & H.	8.3:1
Clawson	11:1
Heintz	6:1 (8.6 Theoretical)
Fairbanks Morse	9.75:1
Ricardo E65	7.0:1
Jessel	8:1
Cranfield-Kushul	6:1 & 16.4:1
Kushul	11.5:1 (overall)
Ricardo E30	4.9:1

The NiCE engine did not suffer from knock, but the compression ratio was relatively low and no attempt was made to exploit a higher ratio. The depth and angle of the wedge combustion chamber were considered to greatly affect the engine performance and the use of over square or large bore configurations would be advantageous.

The YOCP engine operated with a compression ratio of 10:1 and required an optimum ignition timing of 12° BTDC at full load 2000 rpm. A higher ratio of 10.75:1 was also tested satisfactorily with only a limiting of the amount of ignition advance before detonation occurred. The K. & H. engine proved to be knock limited at 5° BTDC ignition advance.

The Clawson was also knock limited at 5° BTDC ignition timing, but the fuel used was JP-4. The Heintz engine operated on regular grade fuel with no detonation problems but again its compression ratio was fairly low.

The ability of the YOCP engine to operate at the high compression without detonation is probably due to the fact that injection was still occurring after the start of combustion and this did not allow time for an end gas zone to form.

The Ricardo E65 engine used a compression ratio of 7:1 in the interests of durability as the engine was intended for an aircraft application, but in development ratios of up to 8.4 had been evaluated.

In the Kushul system the ideal compression ratio for the air cylinder is infinite but mechanical limitations result in a finite ratio having to be used. For the Russian engine this value was not stated although the overall compression ratio was 11.5:1. In the Cranfield-Kushul engine the mixture cylinder had a ratio of 6.1:1 and the air cylinder a ratio of 16.4:1, giving an overall ratio of 7.9:1. If mixture can be prevented from entering the air cylinder no problems with detonation will occur, and Kushul found that it was possible to set the parameters such that a range of fuels could be accepted without problems.

Throttling

The degree of throttling varied widely amongst the engines in this category and could be summarised as follows:-

Unthrottled at any condition;	K. & H.
	Clawson
	Ricardo E65
	Ricardo APC-1
Throttled at idle and light load;	YOCP
	Heintz
Throttled across speed and load range;	NICE
	Fairbanks Morse
	Jessel
	Kushul
	Cranfield-Kushul
	Ricardo E30

Of the completely unthrottled group only the Clawson engine has been utilised in a vehicle, indicating an acceptable degree of combustion control. The K. & H. engine has only been operated at steady states at one speed. The Ricardo engine especially E65 noted the problem of erratic operation between 15 and 40% full load which could have been improved by the use of throttling if a full operational speed range had been required.

The YOCP and Heintz were both throttled at idle and light loads in the interest of maintaining regular combustion. In the case of the YOCP engine it was claimed that throttling at light load was beneficial as the less dense charge in the crankcase reduced the work done by the piston on the crankcase charge. However the throttling effect would increase the amount of residuals retained within the cylinder which reduces the specific heat of the total charge. Whether the reduction in pumping losses outweighed the reduction in specific heats was not quantified. The Kushul engine could be operated with the air cylinder unthrottled, but usually both cylinders were throttled, with a difference in throttle angle between them.

Both the NICE and YOCP engine were shown to be capable of operating at air fuel ratios of 20/21:1 at idle, providing throttling was used.

Rig Tests

To understand the air motion in the piston port controlled engine a rig was used from which the NICE engine was developed. A glass cylinder liner and acrylic cylinder head were used to allow gas flow to be visualised. Metaldehyde (CH_3CHO), was added to the scavenger gas and by passing a beam of light through the cylinder the path of the tracer gas could be photographed with a high speed camera. It was possible to observe the motion in any plane required by altering the light beam angle. This rig testing allowed the effect of the shape of the combustion chamber, position and size of the scavenging and exhaust ports, scavenging and exhaust pressures, etc. to be determined.

In the development of the YOCP engine an enlarged cylinder head rig was used. This enabled photographs to be taken during injection which led to the optimisation of the direction of injection, quality of fuel spray, direction of air stream and cylinder head cavity shape. These results were applied to the actual engine used for the test bed performance measurements reported later.

Rig tests on a specially built Kushul engine were used to determine the dwell angle between the cylinders, the passage dimensions between the two cylinders, compression ratio and other important parameters before a finalised version was built for evaluation.

Topic Area (b) Combustion Characteristics

No data has been published giving details of studies of the actual mechanism of combustion relating to the systems in this category. Whether true stratification is occurring in some of the systems is open to doubt and several of the two stroke engines have been aimed at overcoming problems with the existing engine types, such as idle and light load operation. From the published data it seems likely that they have improved this shortcoming to an acceptable degree, but the actual method by which this has been achieved is not always clear. In the case of the NiCE and YOCP engines rig tests were carried out as described in topic area (a), but these were not on the actual firing engine. Cylinder pressure diagrams at full load conditions are not available nor is film of the actual combustion process in any of the systems.

Topic Area (c) Performance

The torque curve for the NiCE engine scaled from the published results is shown by fig. 7.17 compared with the diesel engine. The swept volume required for the target horsepower is approximately 40% lower than that required for the diesel engine, but the level of torque produced is much lower. Also the two stroke does not exhibit the typical torque back up of the diesel engine having a relatively low torque curve between 2000 and 4000 rpm. The rapid drop in torque above 4000 rpm is due in part to a limiting of the volumetric efficiency caused by the porting arrangement used. The maximum bmep produced by the NICE engine was 5.5 bar.

The only other reference containing sufficient data to construct a torque curve was for the Kushul and fig. 7.18 shows this. The engine capacity required for the target horsepower is slightly less than for the diesel engine and the torque back-up is similar between 3000 and 4000 rpm, while below this speed the level of torque is lower than for the diesel. The peak bmep produced by the Kushul is 8.4 bar.

The engine friction for the Kushul is shown in fig. 7.19, this being the only data available relating to engine friction in this category. As can be seen the friction is considerably lower than a small high speed diesel engine, mainly due to the lower overall compression ratio used in the Kushul. However the friction is higher than the standard curve for a European gasoline engine, due to the high compression ratio on one bank of cylinders and the losses associated with the throat interconnecting the pairs of cylinders.

No data has been published relating to the effect of turbocharging on any of the systems considered in this category.

Topic Area (d) Fuel Consumption

The fuel consumption of the conventional two stroke is usually inferior to the four stroke, due in part to the charge loss. This is fuel which passes straight through the engine during the scavenge period and is never burnt. This also contributes to the high levels of unburnt hydrocarbon emissions measured in the exhaust. In addition to the fuel which passes through the engine unburnt, there is the fuel lost due to cyclic irregularity at light load operation. Due to poor scavenging or mixture preparation the mixture in the vicinity of the spark plug is not always ignitable and therefore on some strokes either partial or no combustion occurs. This again means that the full energy is not being extracted from the fuel supplied and results in a poor fuel consumption compared with the more controlled situation of a poppet valve four stroke. The two stroke stratified charge engine and exhaust diluent types such as the Jessel use exhaust as the bulk gas instead of air as in the case of the four stroke engine. Therefore it is unlikely that this type of engine can ever have good fuel economy as the exhaust gas reduces the specific heat of the total charge mass.

The stratified charge two stroke systems have attempted to ensure regular combustion at all conditions and to reduce charge loss, mainly by ensuring that fuel is not present during the scavenge period.

Test Bed Results

The figures quoted for test bed results for the NICE, YOCP and Heintz systems are shown in fig. 7.7, 7.8 and 7.9. As would be expected the fuel consumption of the NICE was inferior at all conditions to the diesel engine results which are used for comparison. The minimum consumption recorded was 330 gm/kw-hr at 3000 rpm. This engine is operating throttled, both intake and exhaust, and the compression ratio was only 6.3:1. Although charge loss had been reduced by the addition of the exhaust throttle it was likely that some loss was still occurring as indicated by the very high levels of hydrocarbons in the exhaust. The presence of lubricating oil in the exhaust would also contribute to the hydrocarbon emissions.

The YOCP engine which operates unthrottled except at light loads shows a much better fuel consumption at 2000 rpm, the only condition for which data is available. The minimum consumption is approaching that of the diesel and at light loads is superior. However this engine was not optimised to operate across the speed range and whether this level of performance could be maintained is not clear. The use of the direct fuel injection and a high compression ratio has obviously reduced fuel wastage due to charge loss and cyclic irregularity and indicates that there is a possibility of obtaining much improved fuel consumptions from the two stroke if some extra complication can be accepted.

The Heintz system however, which again was only tested at 2000 rpm does not show a good consumption at all and the minimum consumption is only 350 gm/kW-hr. This system suffered from the lack of a suitable fuel injection system and it was known that considerable charge loss occurred immediately the inlet valve was opened, due to the build up of fuel in the inlet port. This was obviously having a detrimental effect on fuel economy and it would appear that a considerable amount of work is required before this sytem is fully developed.

The only other engines in this category that fuel consumption figures are available for, are the Kushul engines. The Cranfield-Kushul engine is at an early stage of development, but the figures published at 2000 rpm show, fig. 7.10 that in the minimum consumption setting it is equal to the diesel engine baseline at light load. The Kushul engine, fig. 7.11 was thoroughly tested in its final form, following rig tests to achieve the optimum build, and shows a fuel consumption which is superior to the diesel at all conditions between 1000 and 4000 rpm. During these tests the engine was throttled below half power but above this power regulation was by fuel control alone. The air cylinder was also throttled, but by a different amount to the mixture cylinder. In the direct injection form used on the experimental engine, which was the prototype of the engine from which the previous results were obtained, a comparison of performance on gasoline and diesel fuel was measured, see fig. 7.12. The minimum fuel consumption on a gasoline was approximately 250 gm/kW-hr. (compared with 220 gm/kW-hr in final version) and on diesel fuel the minimum consumption was approx. 260 gm/kW-hr. The level of consumption is slightly inferior all across the load range but the engine shows an ability to accept the two fuels with no loss in performance.

Vehicle Results

The NiCE engine is again the only system for which vehicle fuel consumption figures have been published and these are steady state consumptions across the speed range, fig. 7.13. The NiCE engine is compared with a conventional two stroke mounted in a 725 kg (1600 lb) vehicle and shows an advantage of 19% at 30 mph reducing to 9% at 50 mph. These values however are obtained from a very light vehicle, this being the type of vehicle in which these engines are used in Japan. The only vehicle fuel consumption figures available for the Kushul engine are a comment that when the engine was installed in a 'Volga' car the Kushul engine used 25% less fuel than the normal 2.45 litre (149 in^3)4 cylinder engine, but without quoting actual figures.

Topic Area (e) Emissions

The exhaust emission levels from the uncontrolled two stroke engine are typified by very high levels of hydrocarbons and low levels of NO_x. The HC levels are due to charge loss during scavenging and incomplete combustion in addition to lubricating oil added to the in-going fuel. The low NO_x levels are due largely to exhaust dilution of the incoming charge causing an internal exhaust gas recirculation effect. The method of analysis used for measuring the hydrocarbon emissions from two strokes can have a considerable effect on the values and much of the test data reported was measured with non-dispersive infra-red analysers. Although a factor can be applied to convert this figure to that which would have been recorded with a flame ionisation detector this conversion is open to criticism and can have a significant effect on the results. Much of the emission measurements have been made on test bed engines only and in the case of the Ricardo engines no measurements were recorded at all, as at the time the development was being carried out, there was little interest in exhaust emissions.

HC, CO & NO_x from Engine Test Beds

The NICE engine emission levels at steady states are shown in fig. 7.7 and indicate that in spite of the improvements in cyclic irregularity of combustion control the level of HC is still very high compared with the diesel baseline engine. It appears that charge loss is probably still occurring at light loads causing the very high levels of HC which were recorded. In addition the high CO levels suggest that combustion is incomplete at all conditions which will also aggravate the HC problem. The very low NO_x emissions would be expected as the inefficient and incomplete combustion combined with the low compression ratio would result in low peak cylinder temperatures which would minimise NO_x formation. These levels of exhaust emissions are an indication of the problem that the two stroke has to overcome before it can be considered as an automotive power plant.

The YOCP results however, fig. 7.8 suggest that the form of charge stratification that they have adopted is effective, especially at the lower load conditions. The levels of HC and CO, equal to the diesel engine indicate that at certain conditions the engine is definitely operating in a stratified charge mode. The mixture strength used at the higher loads was rich of stoichiometric and would account for the rapidly increasing CO level and levelling of the NO_x emissions. This system is much more promising as the object of low emissions and fuel consumption is being achieved, especially at light loads, the important area for vehicle operation. However more development is required to prove that this system can be matched all across the speed and load range required for vehicle operation. Some sources indicate that even with direct fuel injection on the two stroke some form of after-burning device, possibly a thermal reactor will be required to reduce HC levels sufficiently for future legislative requirements. It was noted that although the YOCP engine added oil to the in-going air the exhaust gas claimed to be clear of smoke under all conditions. During the tests it was noticed that the exhaust had a pungent odour which was suspected to be formaldehyde.

Emission results from the limited data published on Kataoka and Hirako engines are shown in fig. 7.14 and indicate very high levels of HC especially at low loads. The CO levels are also high whereas the NO_x levels are very low. This engine appears to require a considerable amount of development before its optimum conditions are achieved as the geometry of the pre-chamber and throat require intensive development to give the best results. These results compared with the NICE engine indicate that the addition of a pre-chamber and direct injection is not in itself the answer to the emission problem with the two stroke engine, as the HC emission levels are no better than those obtained using a carburetted mixture fed into the crankcase.

Test bed figures are not available for the Clawson engine, and the only data relating to the Heintz suggests that the HC emissions are very sensitive to air/fuel ratio and timing of the inlet valve. An increase of HC emissions with air/fuel ratio and a reduction as the speed increased was noted. As had been mentioned previously the fuel injection system used on the Heintz engine was probably responsible for many of the problems with fuel consumption and HC emissions and this trend may be different if a superior injection system was used.

The Jessel engine was only operated at a single steady state of 450 rpm, 25% full load and the results measured showed very low levels of NO_x, moderate levels of CO and high levels of HC in the exhaust. These values were compared with predicted results and suggested that the intention of achieving a thermal reactor effect within the cylinder was not being achieved and that a loss of stratification was occurring during combustion. Instead of the exhaust residuals remaining in one zone and being heated by the combustion of the fresh charge, it seems likely that the fresh charge expanded into the residuals during combustion and was quenched and diluted resulting in a cylinder full of mixed partially combusted residuals. Therefore the idea of recirculating the freshly combusted portion and exhausting the recombusted residuals was not being achieved and the gas exhausted contained a proportion of the fresh mixture charge which had not been burnt completely. Had the thermal reactor effect been achieved by maintaining charge stratification at all times very low levels of both CO and HC and low levels of NO_x would have been expected.

At the time of the development of the original Kushul engine no emission measurements were made. However some limited data is available from the Cranfield-Kushul engine at one speed only, see fig. 7.10. These results indicate that the conditions for minimum fuel consumption are not the same as for minimum HC's, but that in both conditions these are considerably higher than a diesel engine. The CO level however is very comparative with the diesel elvels as are the NO_x levels. The minimum HC levels are achieved at present by operating at an overall air fuel ratio of stoichiometric and throttling to achieve the required load. The high HC level was thought to be a result of mixture flowing into the air cylinder during the compression stroke causing a very lean mixture. Most of this mixture was then transferred to the mixture cylinder, but a small amount of mixture remained in the clearance volume. If the clearance volume was small enough it is likely that combustion would not be able to occur, in the same manner as in the squish areas in a conventional engine, and this unburnt mixture would then be exhausted on the next stroke. It was also possible that the cool air injected into the burning mixture may cause local quenching of the flame which could also increase HC levels. The ingress of mixture into the air cylinder is undesirable as Kushul indicated as this can lead to detonation problems and the phasing between the two pistons, inter-connecting throat dimensions and compression ratios used are all important in achieving this. It is possible therefore that reduced HC emission levels could be achieved from the Cranfield-Kushul engine with further development, as this particular engine is at a very early stage of its development.

HC, CO and NO_x from Vehicle Tests

Baseline results from the NiCE engine fitted into a 725 kg (1600 lb) vehicle and tested to the CVS-1 procedure showed that the levels of HC and CO were well outside the primary emission targets of this survey, see fig. 7.15. The NO_x levels however were equal to the secondary target. To reduce HC and CO emission levels the engine was equipped with an afterburner which successfully reduced HC, CO & NO_x within the secondary emission targets, see fig. 7.15. Unfortunately no data relating to the fuel consumption during these tests was published. The use of the afterburner reduced the HC emissions from 18 to 0.03 gm/mile, the CO from 21 to 1.56 gm/mile and the NO_x from 0.4 to 0.11 gm/mile. The afterburner used to achieve this large reduction in exhaust emissions was exceptionally complex and would seriously affect the total vehicle fuel consumption. It consisted of a combustor into which the exhaust was fed. An electric air pump, blower and burner were used to further oxidise the HC and CO. The burner was ignited with a glow plug and used a separate fuel supply and the device was similar to a gas turbine combustor. It is stated that one of the problems with the device was to find a method of effectively utilizing the heat that it produced. Although obviously very effective it appears a completely impractical and inefficient device to consider for use on a vehicle engine.

The Clawson concept was applied to a 3 cylinder SAAB two stroke engine and tested in a 1000 kg (2250 lb) SAAB vehicle. These emission results to the CVS-2 procedure published by E.P.A. are shown in fig. 7.16 and indicate that although the NO_x was within the secondary emission target and the CO very close to it the HC levels were 6.4 gm/mile. This figure was considered to be approximately 30% low as the use of JP-4 as the fuel for the test meant that many heavy hydrocarbons condensed in the sample line to the flame ionization analyser. Steady state emission tests were carried out to prove this point, but as the present regulations stand, heated sample lines are not required for HC measurement from light duty gasoline engined vehicles. The vehicle was fitted with an Engelhard PTX-4 catalytic converter for HC & CO reduction. It was likely that the amount of excess O_2 present in the exhaust was insufficient to allow efficient conversion across the catalyst as no air pump was fitted. Also the exceptionally high levels of HC produced by the two stroke would require a very efficient catalyst to achieve the levels of HC required by the primary and secondary emission targets. As the load was controlled entirely by fuel quantity, the exhaust temperatures would be low and this also would affect the efficiency of the catalytic converter. It is possible that lower HC levels could be achieved if normal gasoline was used and extra air was available for the catalyst, but it is also likely that further development was required on the combustion chamber and injection system matching to reduce baseline HC levels.

Topic Area (f) Multi-fuel Capability

None of the engines in this category have been designed specifically for multi-fuel operation, but in some cases due to the method of mixture preparation used, the ability to accept a wide range of fuels has occurred.

No data is available for the NiCE engine and from the understanding of the combustion system it is unlikely to accept a wide range of fuels although due to the low compression ratio, low octane gasoline probably could be accepted.

The YOCP engine has been operated on kerosene at a reduced compression ratio of 7.2:1; cold starting and normal operation were achieved without any problems, but an increase in fuel consumption was noted. It would be expected that the YOCP system should have a wide range of fuel acceptance as the combustion process reduces the possibility of end gas formation.

The Clawson engine, a modified 3 cylinder SAAB unit was operated on JP-4 (Research Octane No. 45, Motor Octane No. 43), during the emission test measurements made by E.P.A. It was considered that it may be beneficial to the HC emissions to use gasoline, as this would reduce the heavy hydrocarbons present and allow the catalytic converter to be more effective. Further work was intended on this engine to evaluate the use of methanol and methanol/gasoline blends. The use of direct injection should make this possible as the problems of achieving vaporisation experienced with a carburettor would be reduced.

The Heintz engine was operated on regular grade gasoline, but no attempt was made to operate on any other type of fuel.

The Kushul system again does not allow the formation of an end gas and the engine should be relatively insensitive to fuel quality. The engine was operated on a fuel of zero octane, a mixture of iso-octane and white spirit without any signs of detonation. Even operation on white spirit alone did not cause detonation, provided that the air cylinder was kept free of mixture. A direct injection version of the engine was operated on diesel fuel and gasoline and although the engine had to be operated on gasoline for the initial start operation on diesel fuel was quite satisfactory. Fig. 7.12 shows the results obtained when operating in the direct injection build and indicate that the factors governing successful combustion are not dependent on the fuel in use. The important factor appears to be the introduction of the fresh air charge from the high pressure cylinder into the combustion chamber at the moment at which the expansion of the combustion is at its maximum.

Topic Area (g) Engine Components

Very few of the engines in this category have reached a stage of development at which the reliability of the components can be accurately assessed. The NICE engine has been installed in a vehicle and used for testing but no durability data has been published. This engine does not have any external drives to components and therefore accurate control of ignition timing is assured, but no details regarding the sensitivity of the system to change are known. The exhaust throttle is a potential problem area as the operation of this is an important part of the combustion control system.

The injection equipment used in most of the systems was standard gasoline engine equipment, and in cases where dual injection was used the phasing was important. It was considered that the performance of the Clawson engine could be improved if better atomisation of the fuel could be obtained.

In the case of the Heintz engine the cam timing to operate the pre-chamber intake valve was important and timed fuel injection was considered to be required to overcome some of the problem of excessive HC emissions. Improved engine scavenging and the use of a variable speed Roots blower were also considered to be areas which needed further development.

The Jessel engine was in a very early stage of development and as reported had only operated at 450 rpm. There appear to be many potential reliability problems with components such as the exhaust flap valve, the intake and recirculation timing valves and the phase change mechanism. All these components would need extensive development before they could be considered practical and the phasing and accurate control of the various parameters could present a significant problem, if the engine was to operate over a useable speed range.

The Kushul engine had been operated in a 6 cylinder form (two vertical banks of three cylinders), for 150 hours at speeds of 2500 to 3000 rpm at 75% load with no problems. In addition 30,000 km of trouble free operation in an automobile have been accummulated. The critical phasing between pistons in this engine is a setting that having been fixed will not alter with wear.

Topic Area (h) Cost

The state of development of the engines in this category makes estimating costs for production impossible. Comments regarding practicability and complexity have already been made in topic area (g).

Topic Area (i) Mathematical Models

Kataoka and Hirako predicted the NO_x levels from their pre-chamber two stroke engine by the use of a mathematical model. The results compare NO_x formation in an open chamber carburetted and a pre-chamber direct injected engine. This indicated a variation in NO_x concentration formation in the various zones of the chamber and predicted a low overall level of NO_x in the divided chamber due to the rich mixture strength in the pre-chamber. The measured pressure diagrams have been used to calculate average temperature and heat release rates again comparing the open and divided chamber engine. In the divided chamber the shape of the temperature, heat release and pressure diagrams were similar across the load range, whereas in the open chamber homogeneous charge the rate of pressure rise, temperature and heat release increase with engine output.

Topic Area (j) Engine Size and Weight

No specific data relating to the engine size and weights was published in the references surveyed. Some of the two stroke engines were obviously smaller and lighter than comparative output four stroke units due to the engine design alone. Some of the modifications required considerably increased the size, weight and complexity of the engines. The requirement of an inlet valve for the pre-chamber on the Heintz engine in addition to the extra height of the combustion chamber/cylinder head arrangement begin to increase the size compared with the simple two stroke. The system applied to the Fairbanks Morse engine did not make any significant difference to the engine size or weight and fitted within the confines of the existing engine configuration.

The Jessel engine was not in a suitable state to make a realistic estimate of its actual size or weight, but it was quite clear that it would probably be larger and heavier than conventional unit because of the recirculation system and phase change mechanisms which were part of the concept. It is possible that some more practical arrangement could be developed, such as the six stroke engine which may not be so much larger than a conventional four stroke.

The Kushul principle limits the cylinder configurations which are suitable for this system and the engine developed was a 6 cylinder unit of 2.75 litres (168 in^3), developing 57 kW (90 bhp) at 4000 rpm. At the time this engine was developed, the early 1960's, this power output was reasonable for this engine capacity 24 kW/litre, and is still equal to automotive diesel standards. The engine unit is fairly compact, being two rows of three cylinders, which with the extra width of the manifolding each side make an engine which is nearly square.

Topic Area (k) Noise and Vibration

The noise of the piston port controlled two stroke engine is very much affected by intake and exhaust silencing, due to the effect of the rapid opening ports. The rate of cylinder pressure rise will also affect this level, but in the conventional two stroke to a lesser degree. The vibration experienced with two strokes is due in part to erratic combustion especially at light loads and idle.

The only engine from which noise measurements had been made in a vehicle was the NICE and this was only of the exhaust noise at idle. The microphone was positioned 1 mitre behind the vehicle, 1.1 metres above the ground. At an idle speed of 900 rpm the measured noise was 55/56.5 dBA compared with 59/62 dBA for the conventional engine in the same body. The vibration at idle and low speeds was also claimed to be far superior to the conventional two stroke, being indistinguishable from a four stroke. The reduced vibration had resulted from improved cyclic regularity and at idle the percentage of cycles on which full combustion occurred was increased from 50% to 100%.

The YOCP engine also showed a great improvement in idling regularity since the low rate of cylinder pressure rise and the small torque fluctuation of the 2 stroke combined to make a smooth and quiet unit.

The pre-chamber type 2 stroke engines are possibly noisier than the open chamber although no definite figures are available to prove this. The Clawson engine was reported to be noisier than a conventional 2 stroke at idle and low loads due to the rapid rise in cylinder pressure.

The Cranfield-Kushul engine was thought to be rough and noisy during the engine tests, but it was suggested that this could be improved by further development, as at present the engine was at an early stage. The Kushul solution to overcome the vibration was the use of two parallel banks of cylinders instead of the in-line arrangement of the Cranfield-Kushul, but a horizontally opposed four cylinder would work equally well.

Cylinder pressure diagrams at full load have not been published for this category of engine as much of the work was aimed at improving light load combustion and the cylinder pressure measurement was concentrated in this area.

Topic Area (l) Durability and Starting

Due to the state of development of most of the engines in this category no details of durability and starting are available and the comments made in topic area (g) are applicable to this section.

Applications to Motorcycles

Since in-cylinder fuel injection is impractical for motorcycles, only the NiCE and Kushul engines would be applicable. The only practical motorcycle configuration for the Kushul process would be a flat four, as has been used by Honda. At the present limited stage of development there does not appear to be any insurmountable difficulties in such a configuration, and since there are indications of superior fuel economy, further research in this area is justified. If the high HC emission problem is soluble, and here the chances of success are better than with most other stratified charge processes, then the Kushul process would become very attractive for lower emission levels than those considered here. At the level of the present proposed motorcycle legislation, the fuel economy advantage of the Kushul process could not outweigh its extra complexity and design constraints, particularly in small engine sizes.

The HC level of the two stroke NiCE engine is very high and certainly greater than four stroke engines. In addition, the emission potential of this engine under transient conditions is unknown.

Conclusions from Category VII

1. Twelve systems were identified in this category covering, two stroke, exhaust diluent and miscellaneous engines. Of these, four had been developed to a multi-cylinder state. One of these, the Fairbanks Morse, is a stationary gas engine which is produced for sale. Of the three remaining systems, the

NICE, Clawson and Kushul, the latter had undergone the most intensive development, durability and road testing.

2. All of the systems applied to the two stroke engine added expense and complexity to the basic cheap and simple engine without overcoming all the short-comings.

3. Throttling was required on many of the systems to achieve operation over a wide speed range, the Clawson engine being the only one to operate unthrottled over the whole load and speed range.

4. The torque curve of the Kushul engine to meet the power requirement was similar to that of a diesel engine and it had lower friction losses.

5. Generally the fuel consumption of all the two stroke engines was poor although that of the YOCP was equal to that of the diesel at one speed only. The Kushul engine fuel consumption was very good, being equal or superior to the baseline diesel engine at all conditions between 1000 and 4000 rpm.

6. The limited information relating to the Kushul engine suggests that the vehicle fuel consumption was 25% lower than with a conventional engine. This figure would mean that under road conditions it was approaching the economy of a diesel engine.

7. The basic exhaust emissions from the two stroke engines show high levels of HC and low levels of NO_x. The limited data from the Cranfield-Kushul engine also suggests a similar trend although it is possible that development will reduce the HC level.

8. Emission tests on a vehicle fitted with a Clawson two stroke engine indicated that with an oxidation catalyst it was not possible to reduce the HC levels to the primary requirement. The only engine to achieve the primary and secondary requirements for HC, CO and NO_x, the NICE, was equipped with a complex afterburner system that could not really be considered practical.

9. The YOCP and Clawson two stroke engines were both capable of operating on fuels such as kerosene and JP-4. However in the case of the YOCP a reduction in compression ratio was required. The Kushul engine did not have an octane requirement at all, provided mixture could be prevented from entering the air cylinder, requiring accurate setting of the various engine variables.

10. With the exception of the Kushul and Fairbanks Morse engines the durability of the systems was difficult to assess as many were still in the experimental stage.

11. The noise levels from the pre-chamber two stroke engines and the Kushul are possible higher than the conventional gasoline engine.

12. It is recommended that the Kushul engine should be explored further. If the HC emission problem can be solved then this engine is a long-term candidate for large motorcycles and automobiles.

Figure 7.1. Ni.C.E. Engine and Kataoka - Hirako Engine.

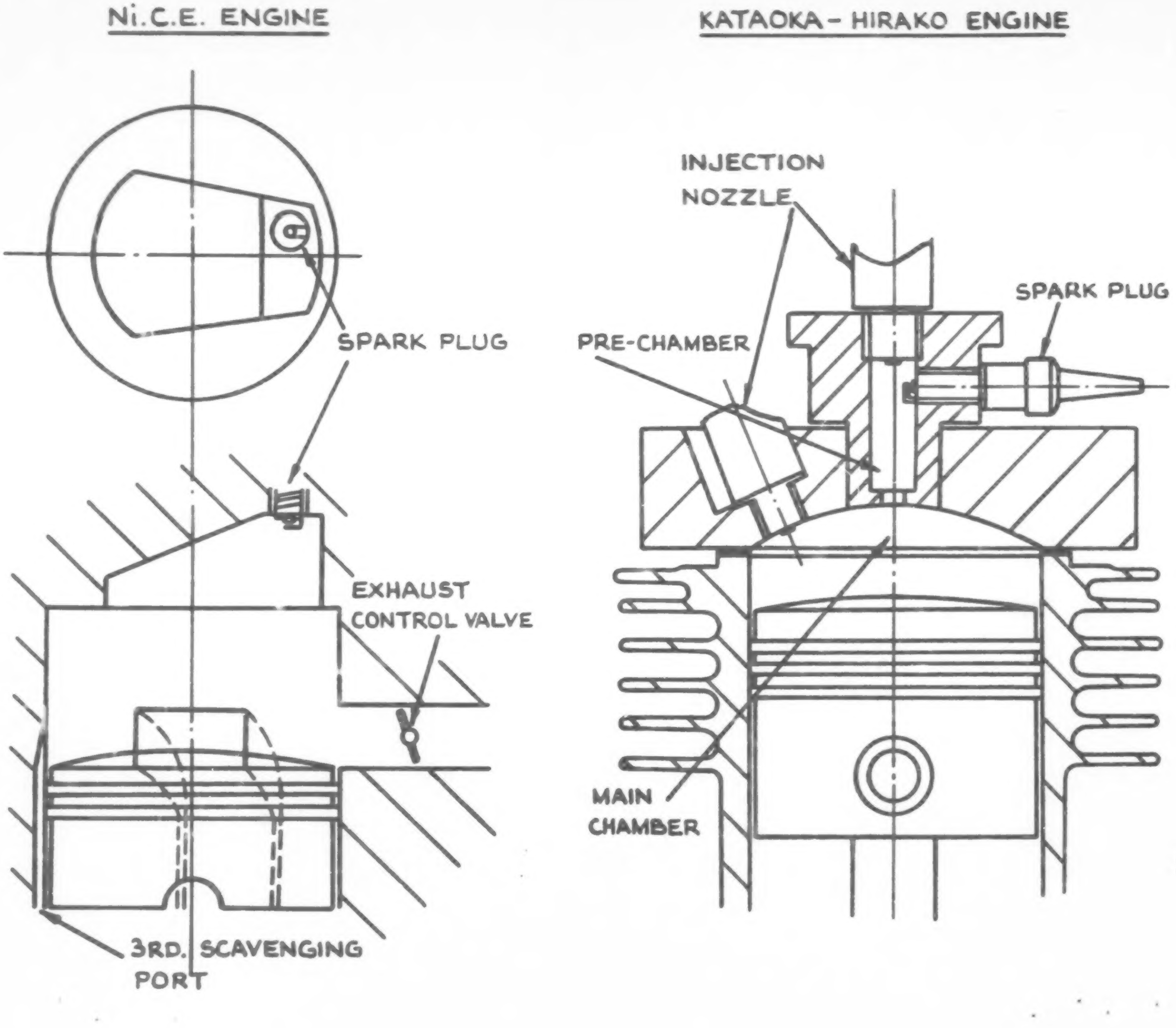

Ni.C.E. ENGINE
SPARK PLUG
EXHAUST
CONTROL VALVE
3RD. SCAVENGING
PORT
KATAOKA - HIRAKO ENGINE
INJECTION
NOZZLE
SPARK PLUG
PRE-CHAMBER
MAIN
CHAMBER

Figure 7.2. Y.O.C.P. Engine and Heintz Engine.

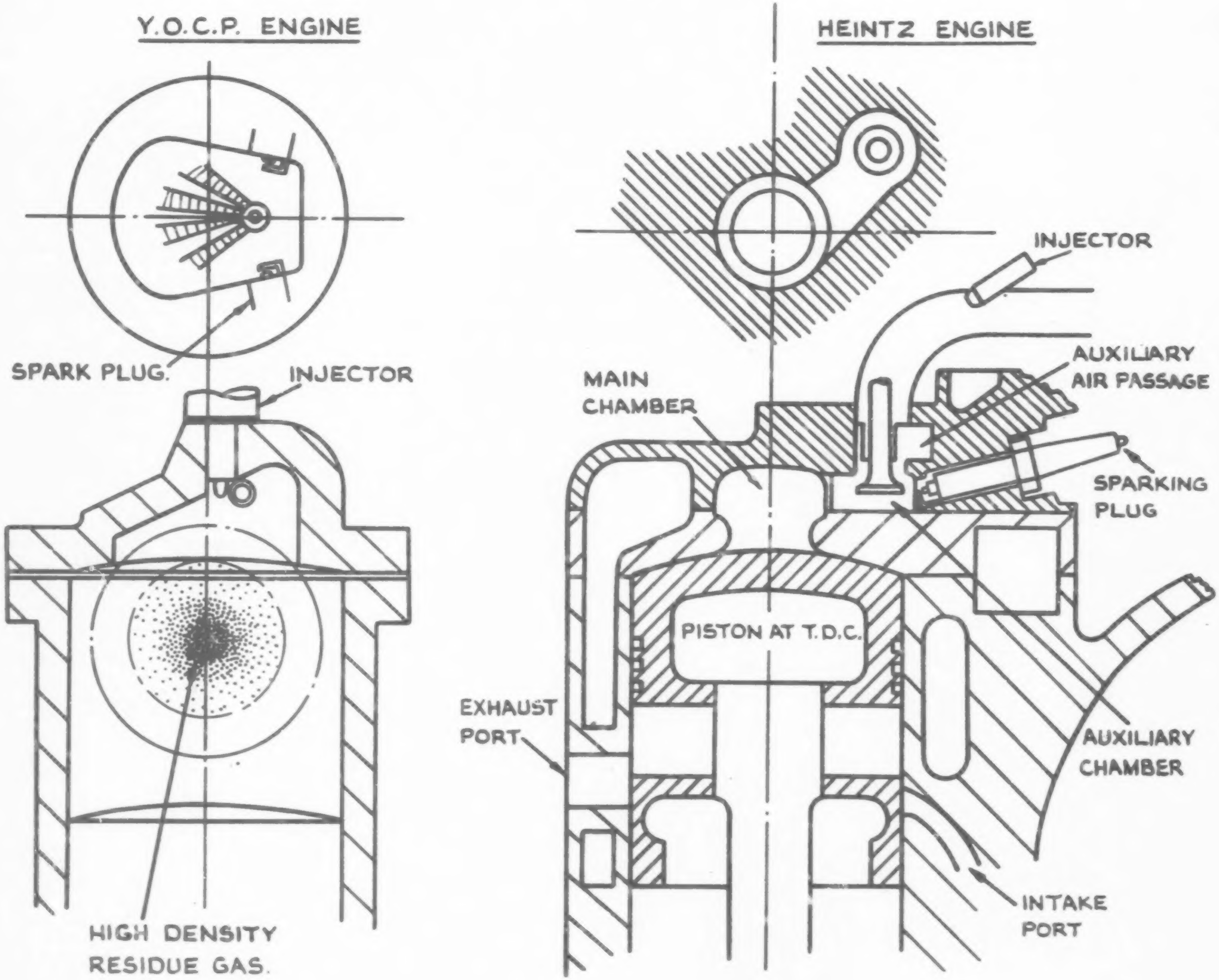
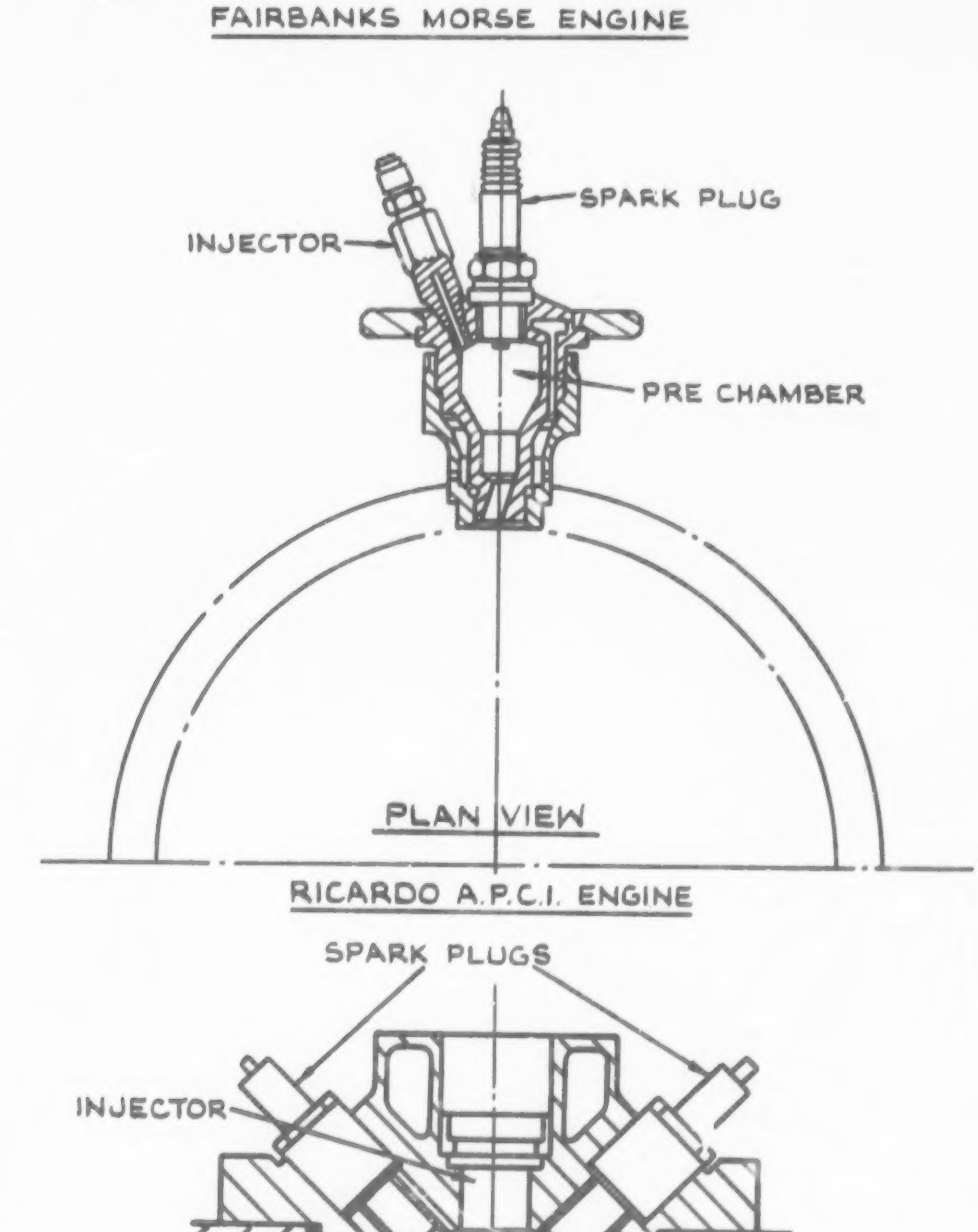

Y.O.C.P. ENGINE
SPARK PLUG.
INJECTOR
HIGH DENSITY RESIDUE GAS.
HEINTZ ENGINE
INJECTOR
AUXILIARY AIR PASSAGE
MAIN CHAMBER
SPARKING PLUG
PISTON AT T.D.C.
EXHAUST PORT
AUXILIARY CHAMBER
INTAKE PORT

Figure 7.3. Fairbanks Morse engine and Ricardo A.P.C.I. engine.


FAIRBANKS MORSE ENGINE

SPARK PLUG
INJECTOR
PRE CHAMBER
PLAN VIEW

RICARDO A.P.C.I. ENGINE

SPARK PLUGS
INJECTOR
PRE CHAMBER
SLEEVE

Figure 7.4. Ricardo E30 Engine.

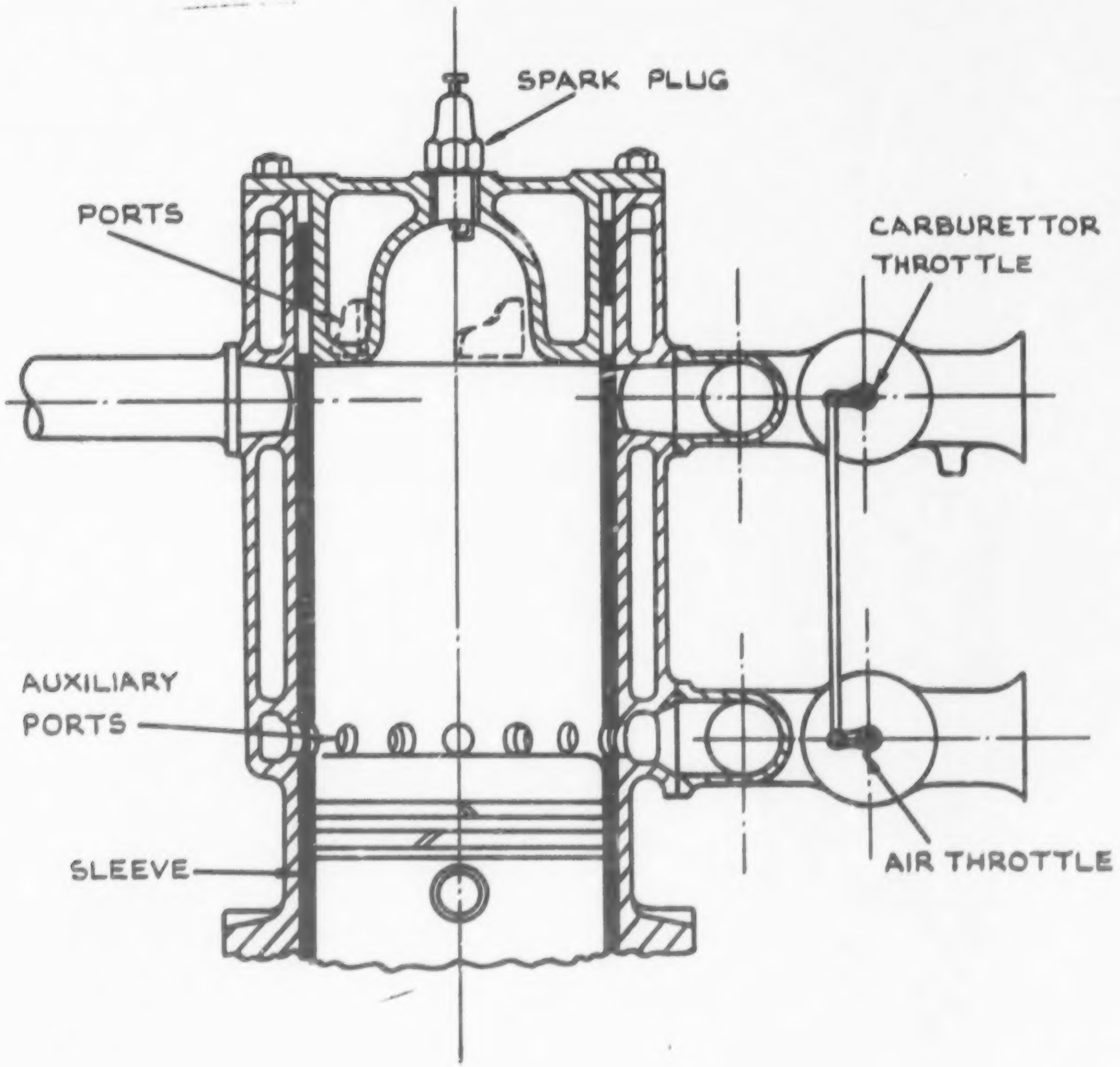
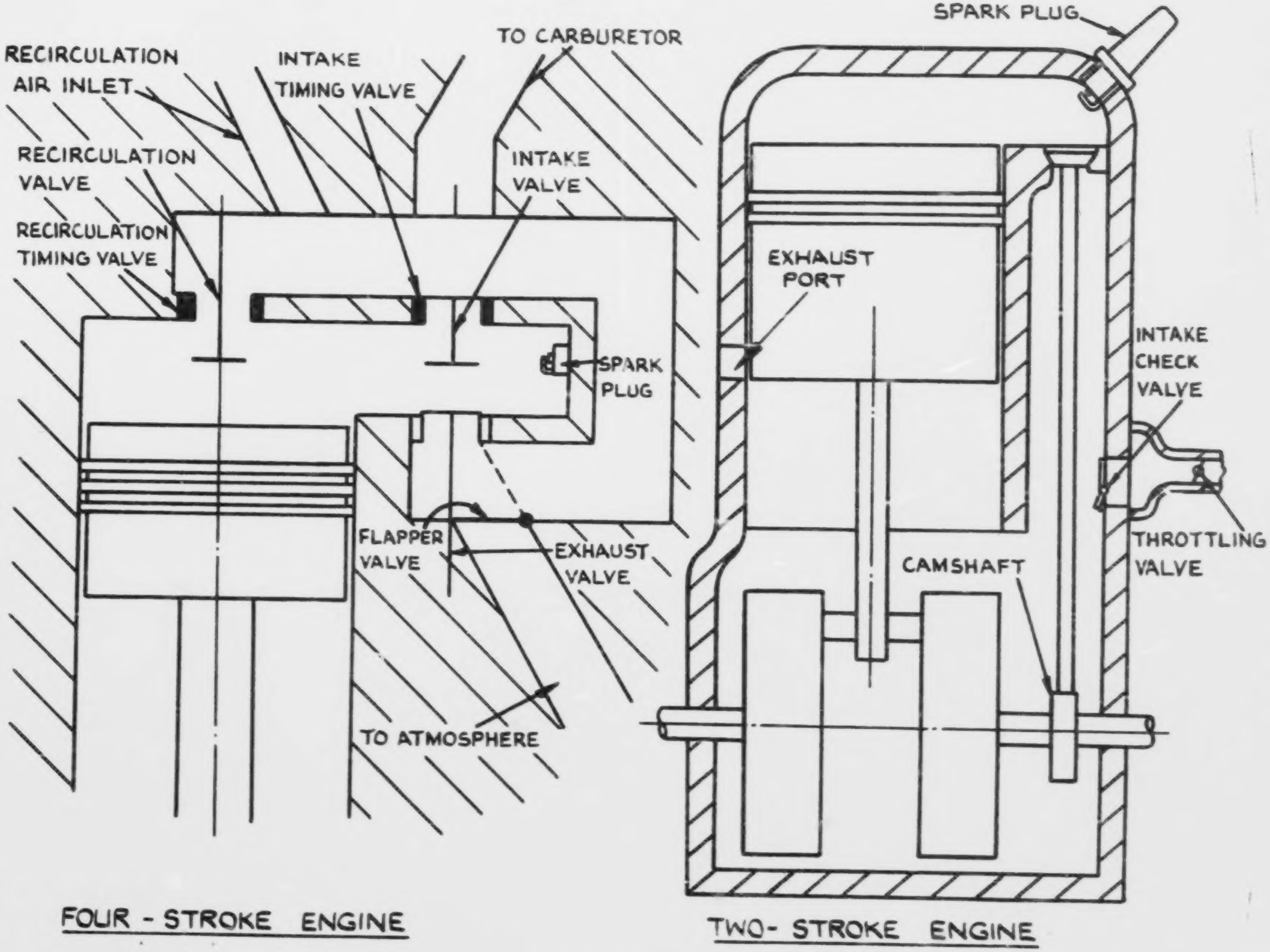

SPARK PLUG
PORTS
CARBURETTOR THROTTLE
AUXILIARY PORTS
SLEEVE
AIR THROTTLE

Figure 7.5. Jessel - Uyehara - Myers Engine.



FOUR - STROKE ENGINE

TWO- STROKE ENGINE

RECIRCULATION AIR INLET
INTAKE TIMING VALVE
TO CARBURETOR
RECIRCULATION VALVE
INTAKE VALVE
RECIRCULATION TIMING VALVE
SPARK PLUG
FLAPPER VALVE
EXHAUST VALVE
TO ATMOSPHERE
SPARK PLUG
EXHAUST PORT
INTAKE CHECK VALVE
THROTTLING VALVE
CAMSHAFT

Figure 7.6. Kushul Engine.

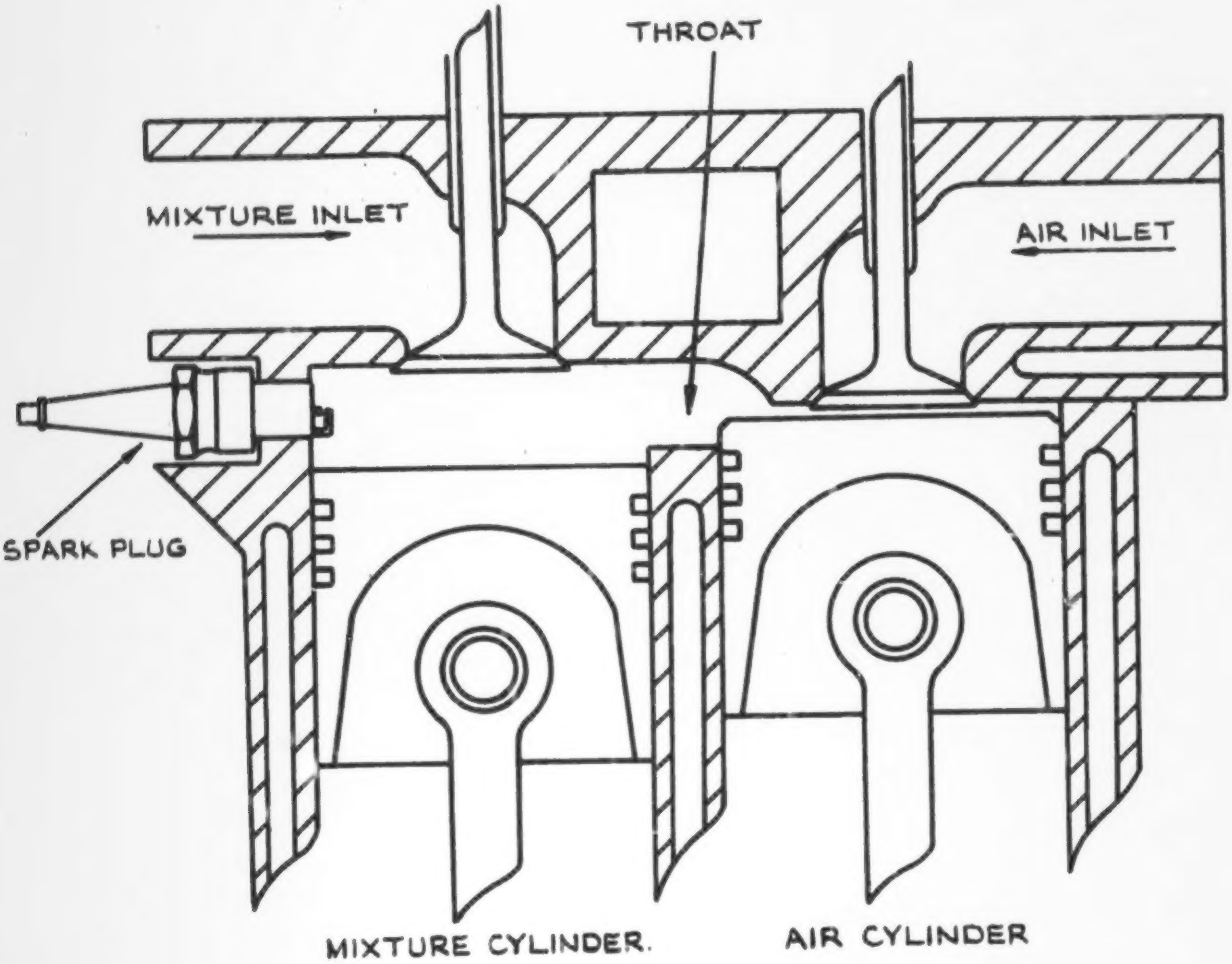

THROAT
MIXTURE INLET
AIR INLET
SPARK PLUG
MIXTURE CYLINDER.
AIR CYLINDER

Figure 7.7. Comparison of emissions and fuel consumption of diesel engine and Ni C.E. two stroke engine extrapolated to target horse power.

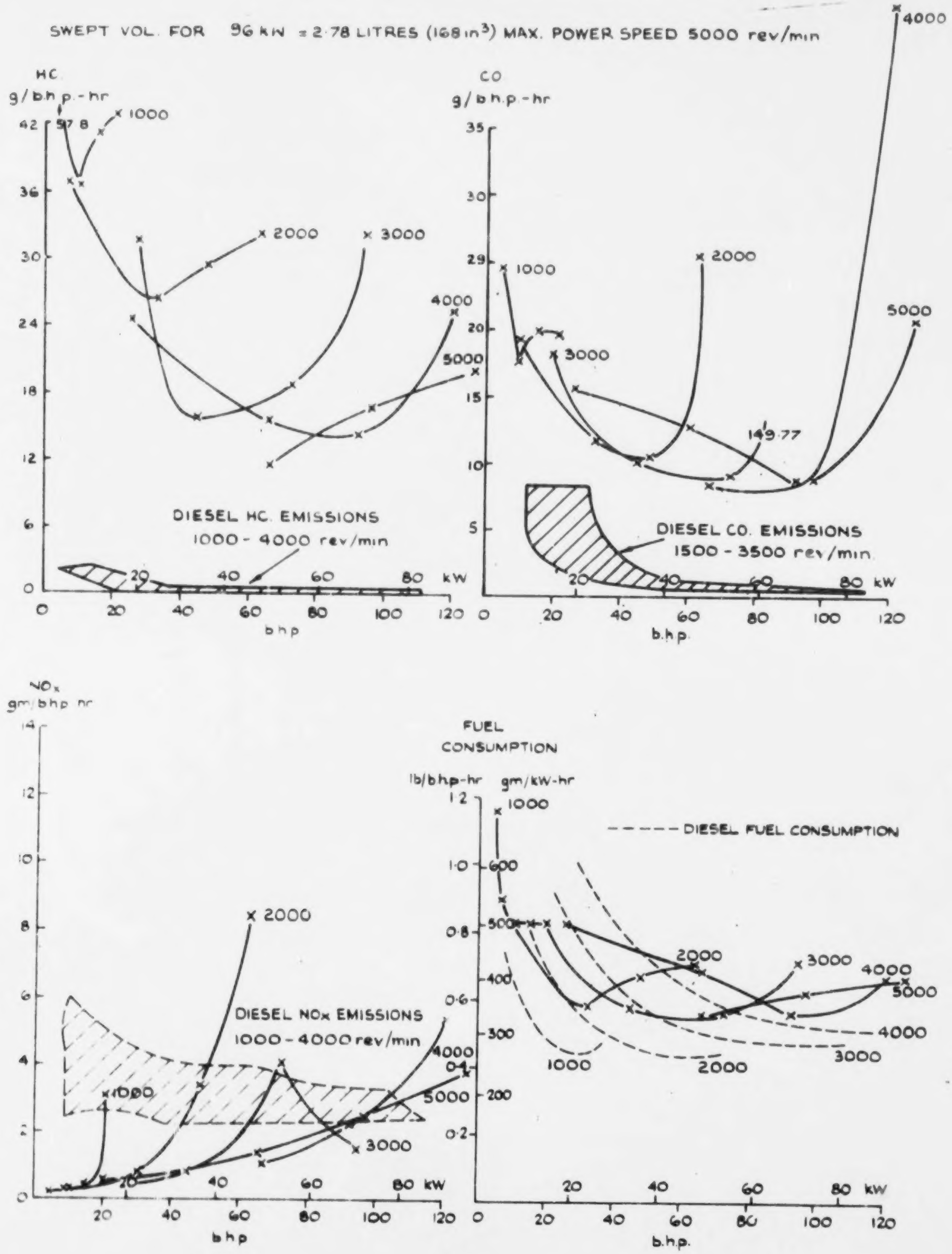


Figure 7.8. Comparison of emissions and fuel consumption of diesel engine and Y.O. C.P. engine extrapolated to target horse power.

SWEPT VOL FOR 96 kW = 2·78 LITRES (168 in³) MAX POWER SPEED 5000 rev/min

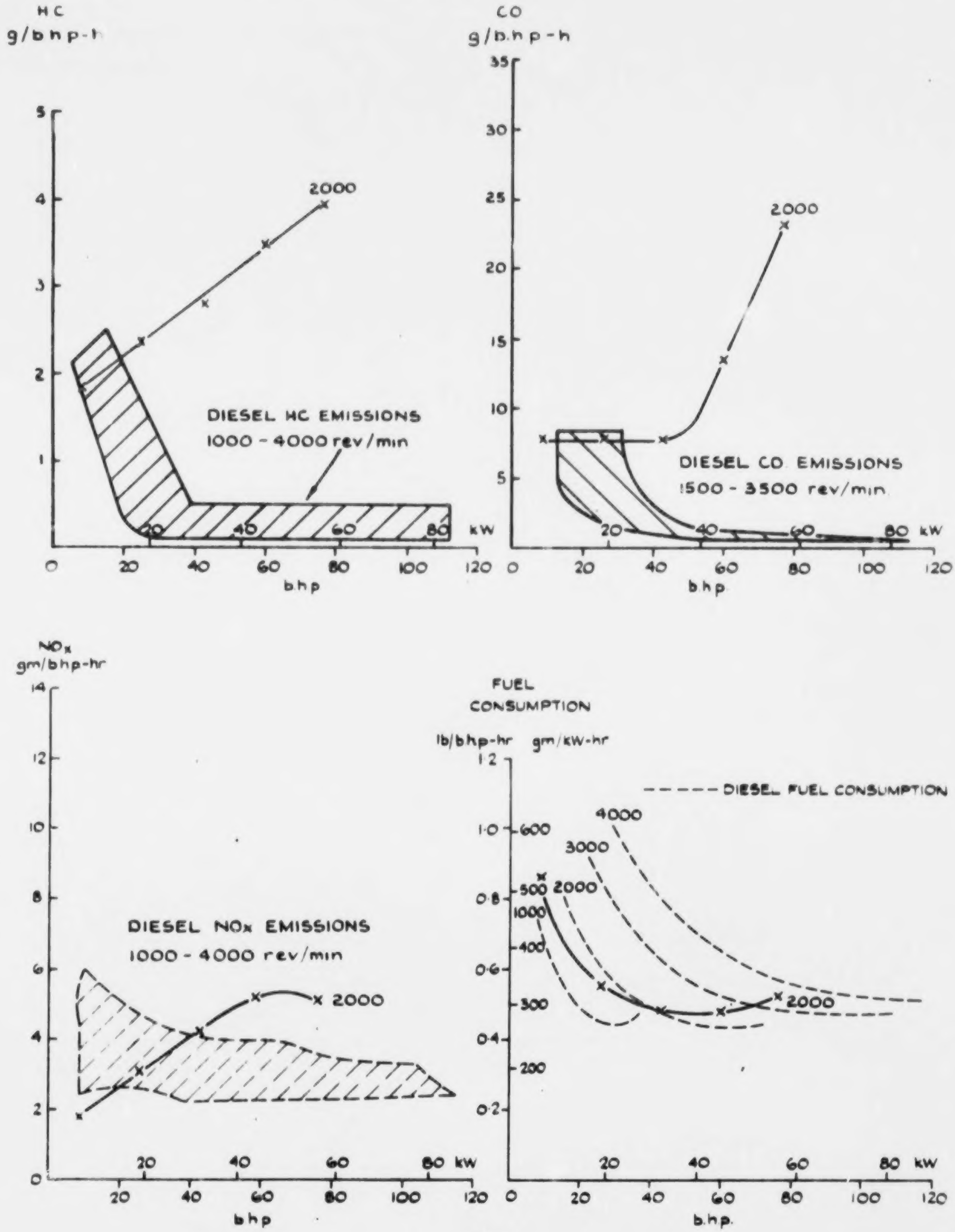


Figure 7.9. Comparison of fuel consumption of diesel engine and Heintz engine extrapolated to target horse power.

SWEPT VOLUME FOR 96 kW = 2·78 LITRES (168in^3). MAX. POWER SPEED 5000 rev/min.

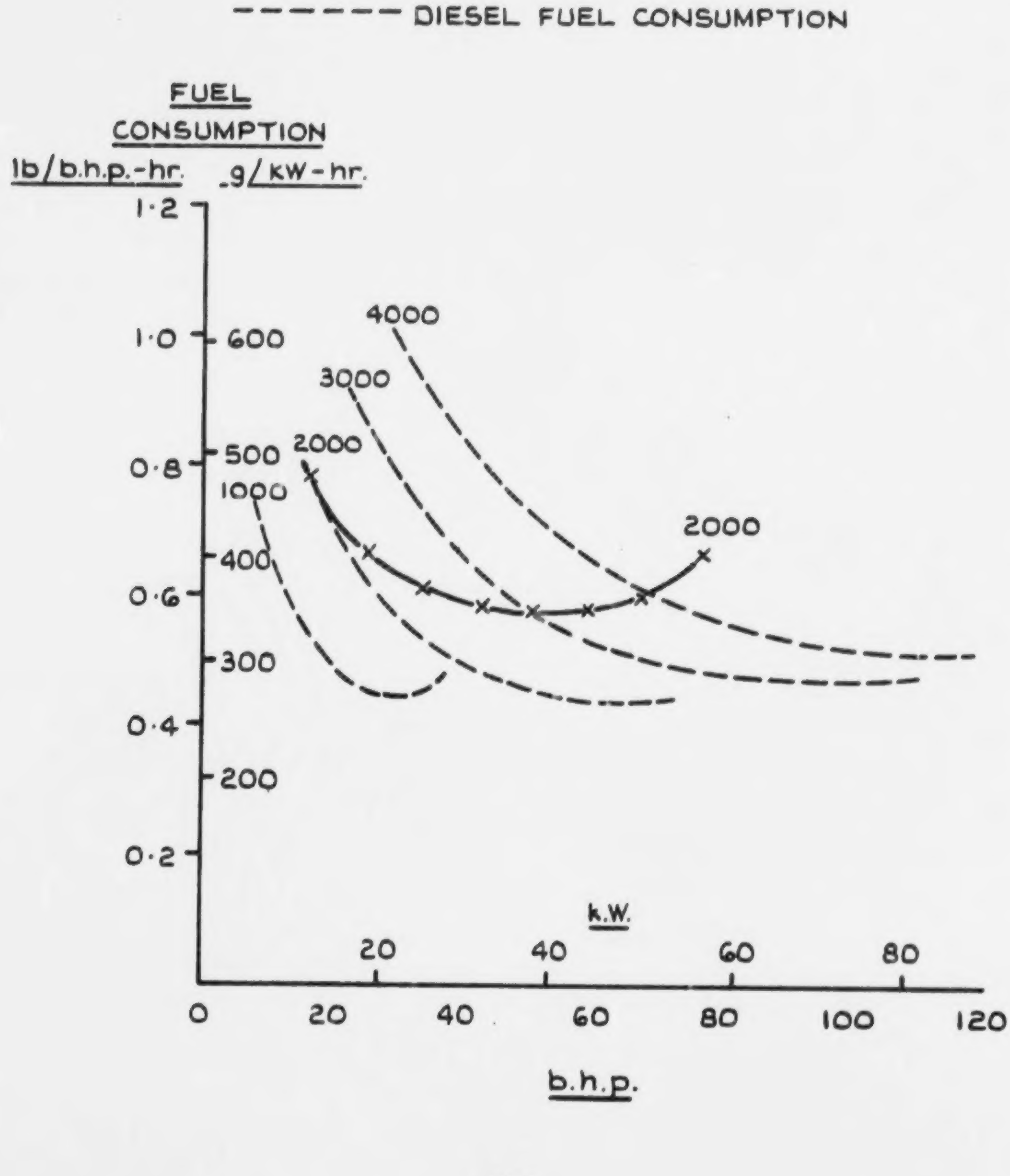


Figure 7.10. Comparison of emissions and fuel consumption of diesel engine and C-K engine extrapolated to target horsepower.

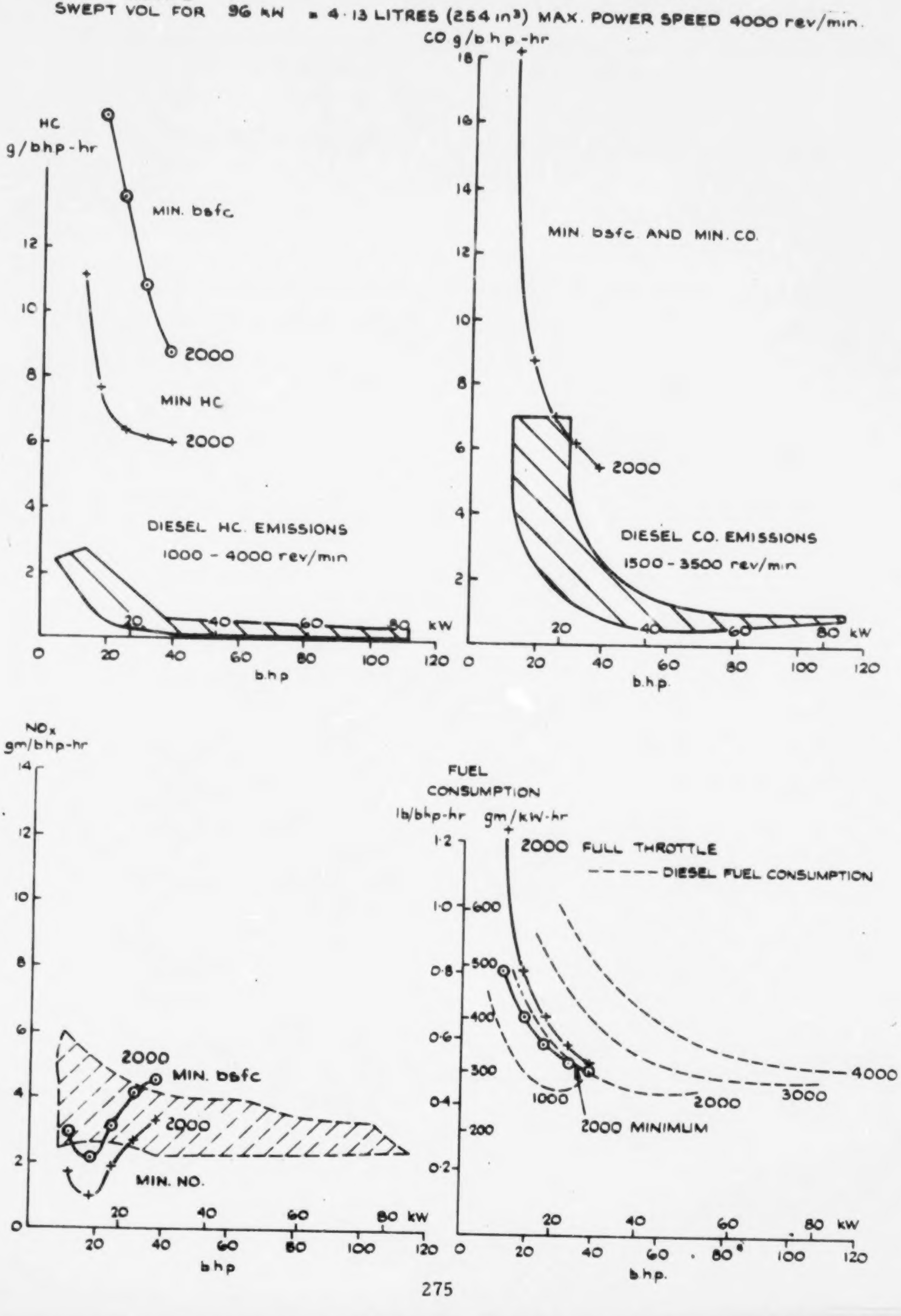


Figure 7.11. Fuel consumption of Kushul engine in comparison with diesel engine extrapolated to target horsepower.

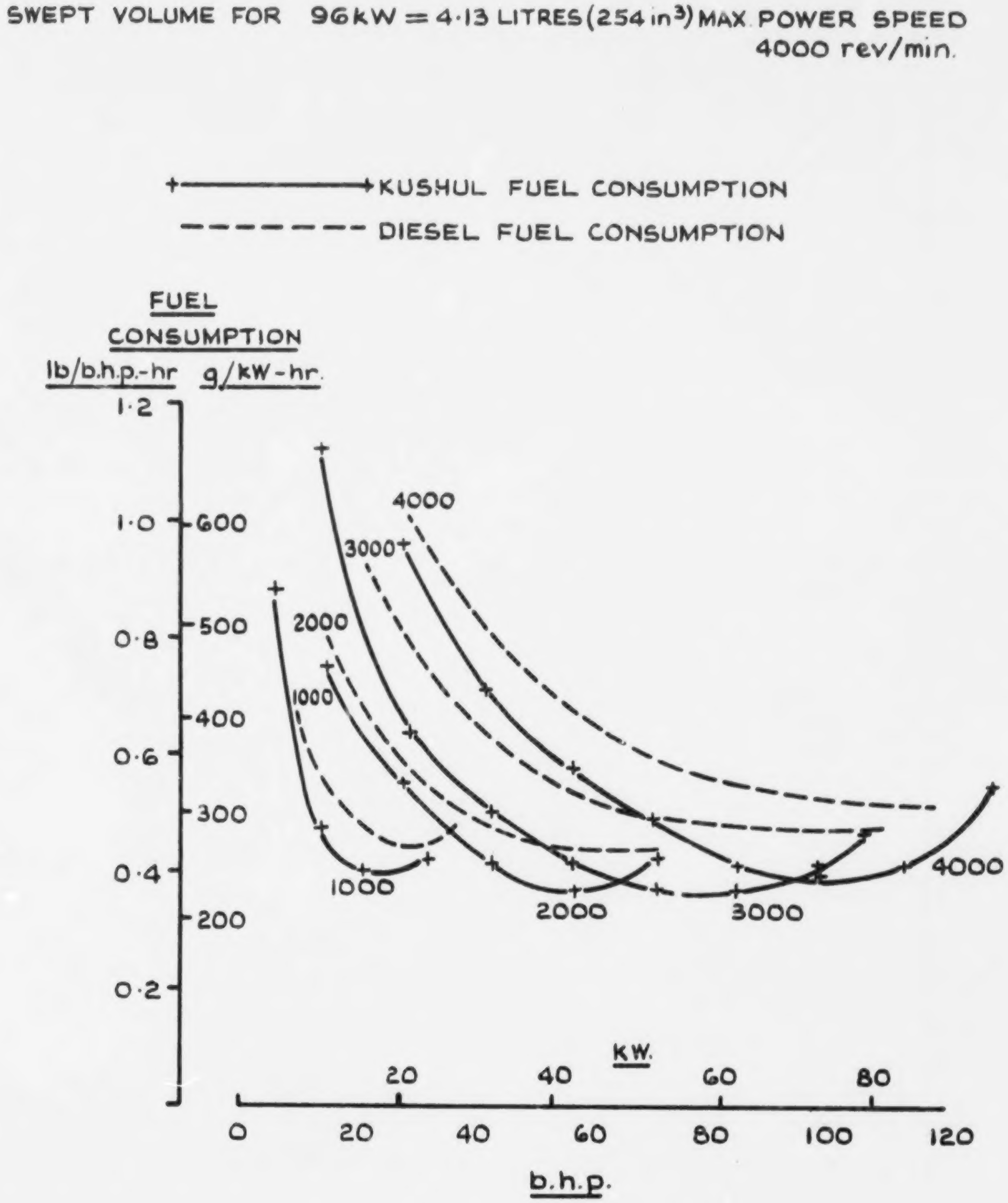


Figure 7.12. Fuel consumption from the Kushul engine with gasoline and diesel fuel.

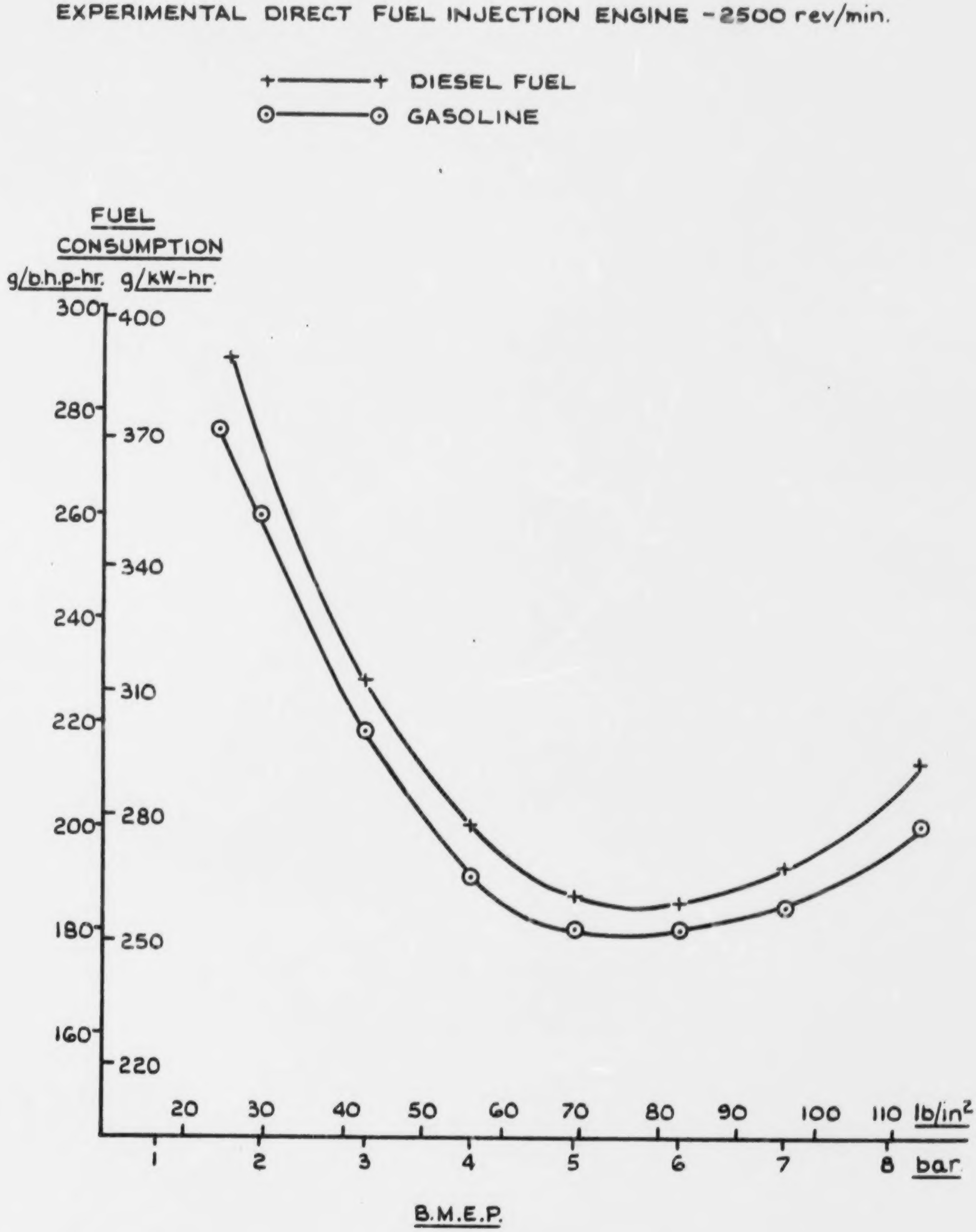


Figure 7.13. Road fuel consumption of Ni. C.E. engined vehicle of 725 kg weight.

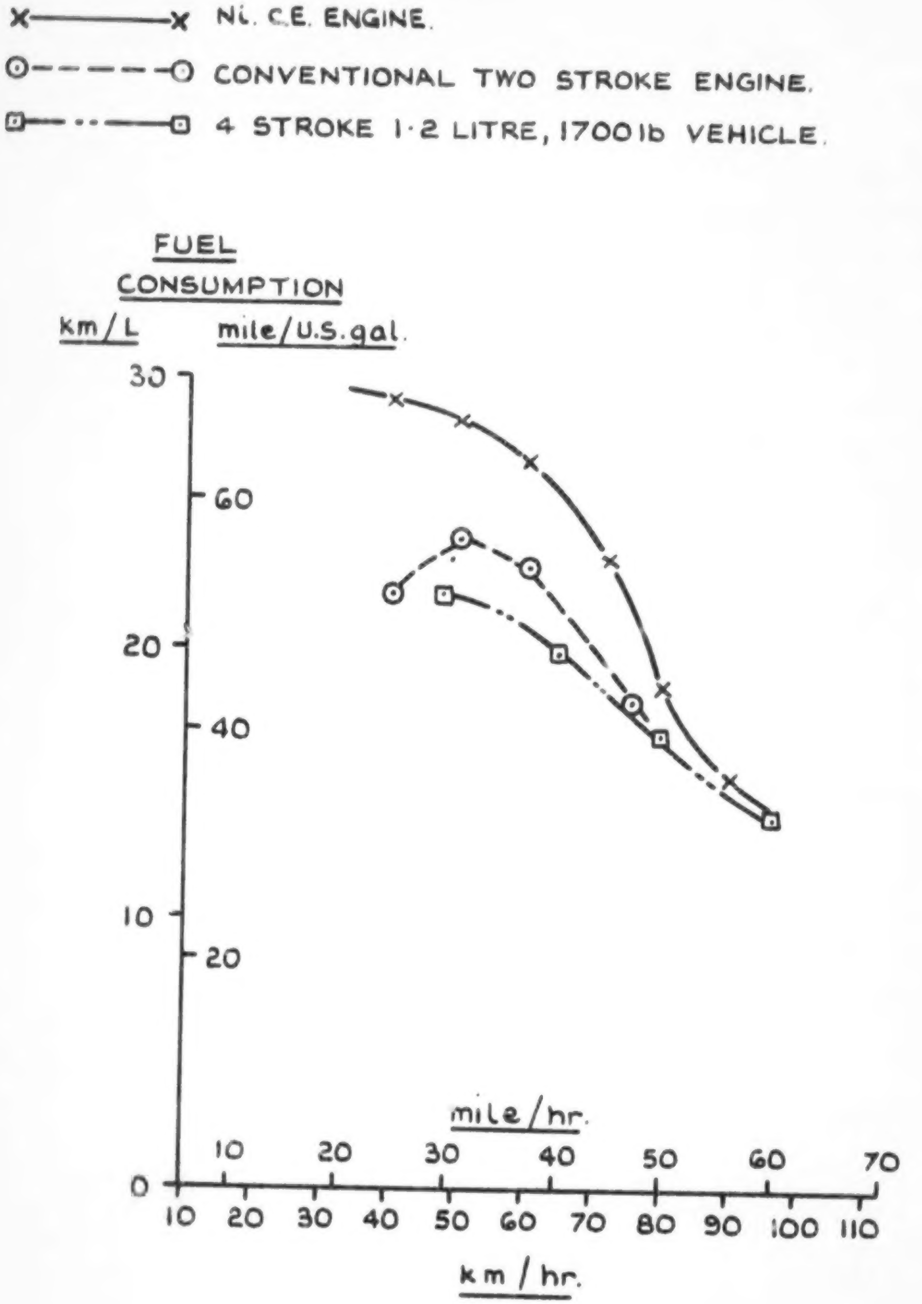


Figure 7.14. Comparison of emissions of diesel engine and K-H engine extrapolated to target horse power.

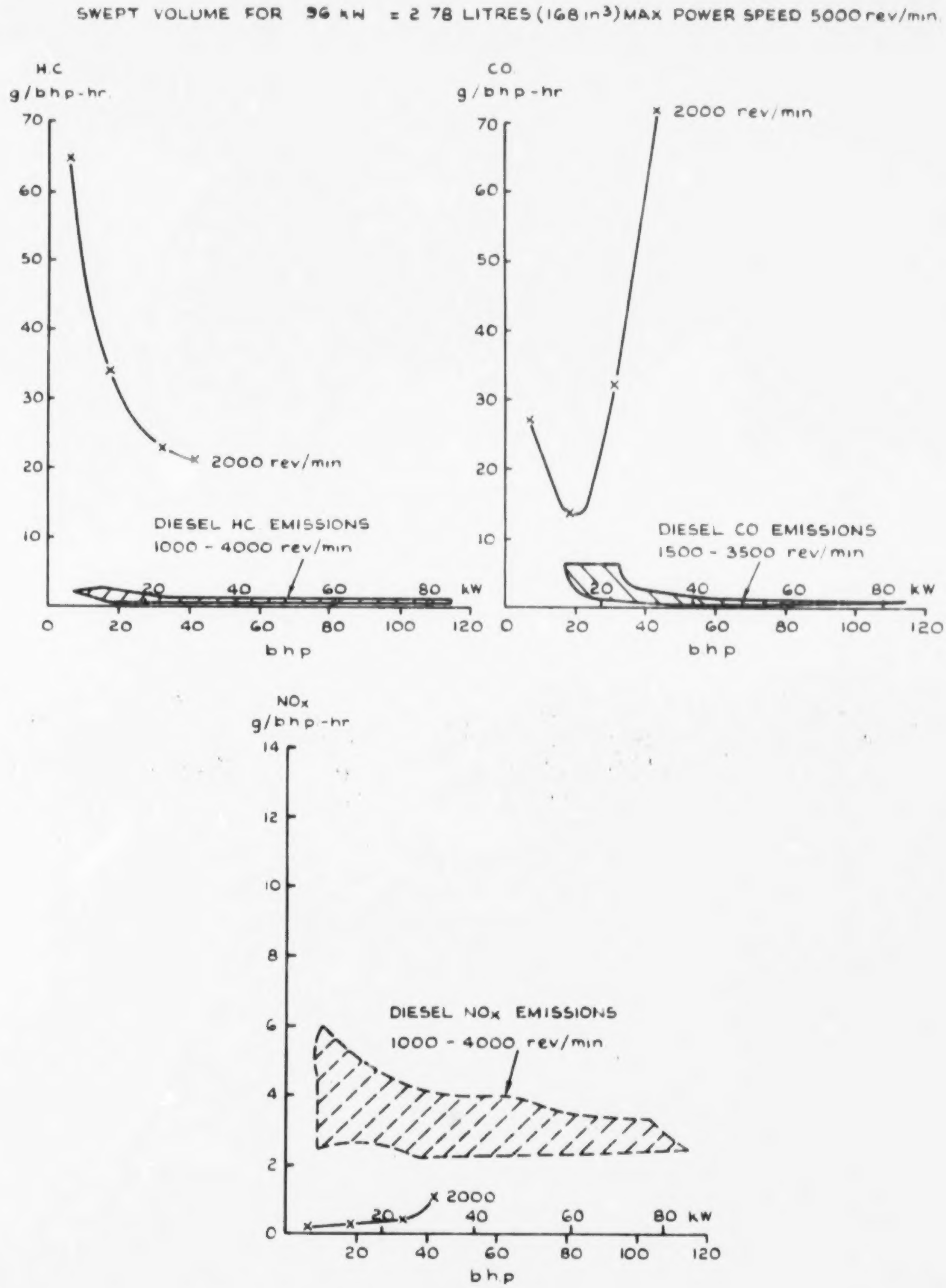
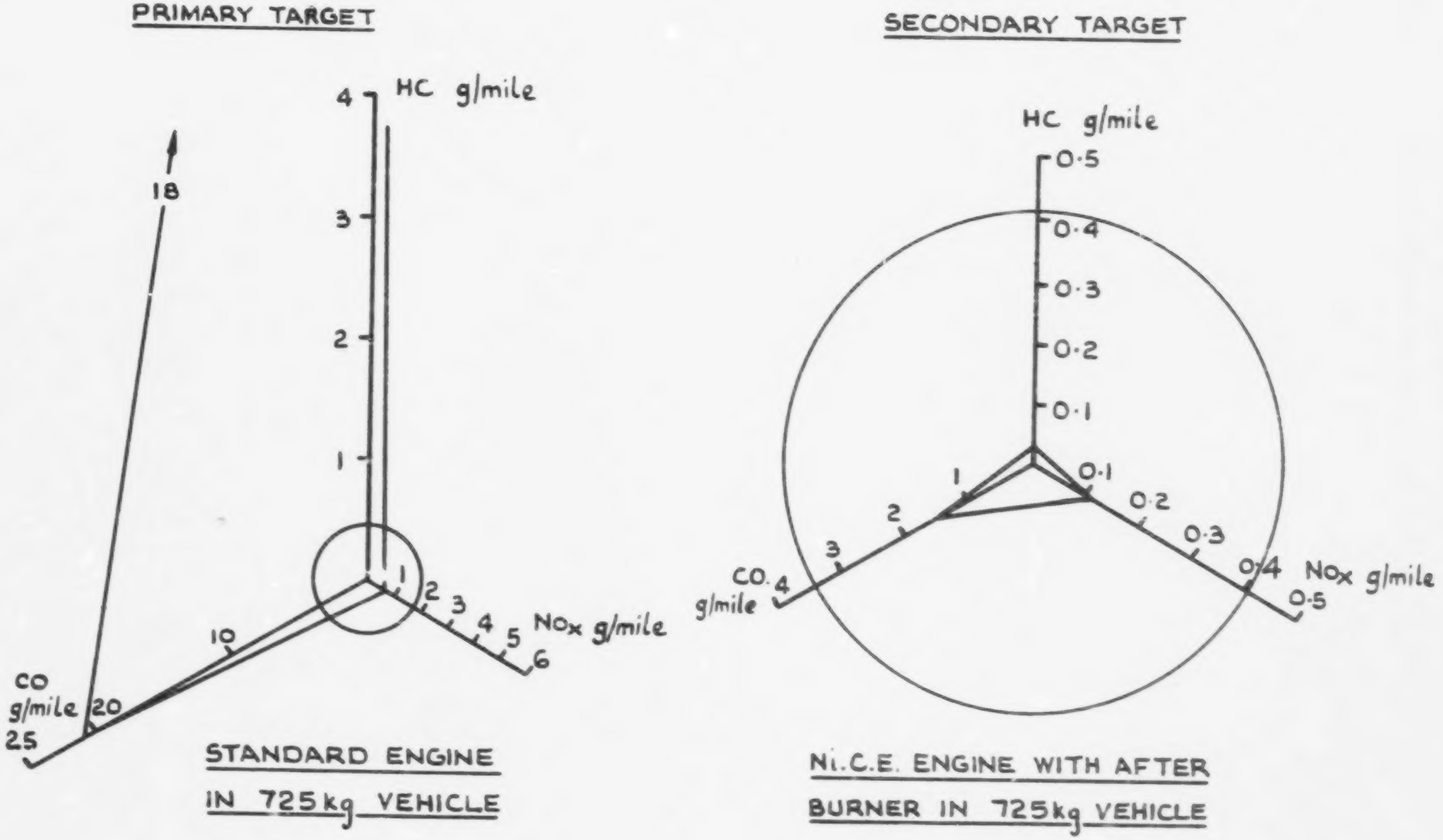


Figure 7.15. Emission test results from Ni. C.E. engined vehicle.



PRIMARY TARGET

STANDARD ENGINE IN 725kg VEHICLE

SECONDARY TARGET

Ni.C.E. ENGINE WITH AFTER BURNER IN 725kg VEHICLE

Figure 7.16. Emission test results from Clawson prechamber engine in SAAB 1000 kg vehicle.

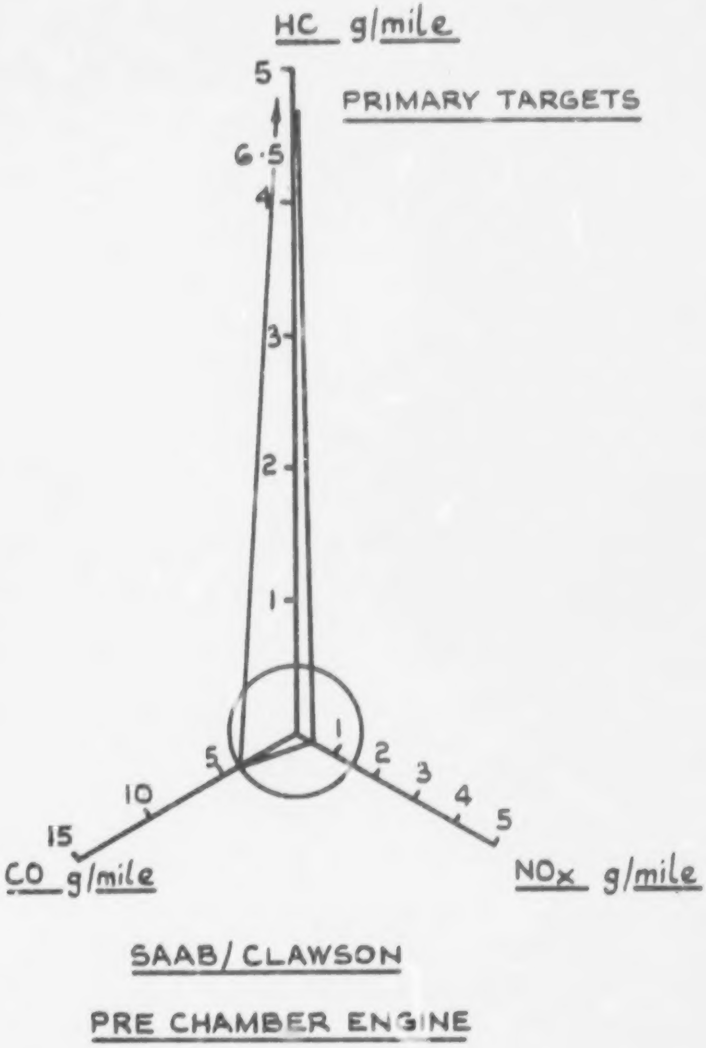


Figure 7.17. Comparison of torque curves of baseline diesel engine and Ni. C.E. two stroke engine.

DIESEL ENGINE

SWEPT VOL. FOR 96kW = 4·77 LITRES(291 in^3)

MAX. POWER SPEED 4000 rev/min.

Ni.C.E. ENGINE.

SWEPT VOL. FOR 96kW = 2·78 LITRES(168 in^3)

MAX. POWER SPEED 5000 rev/min.

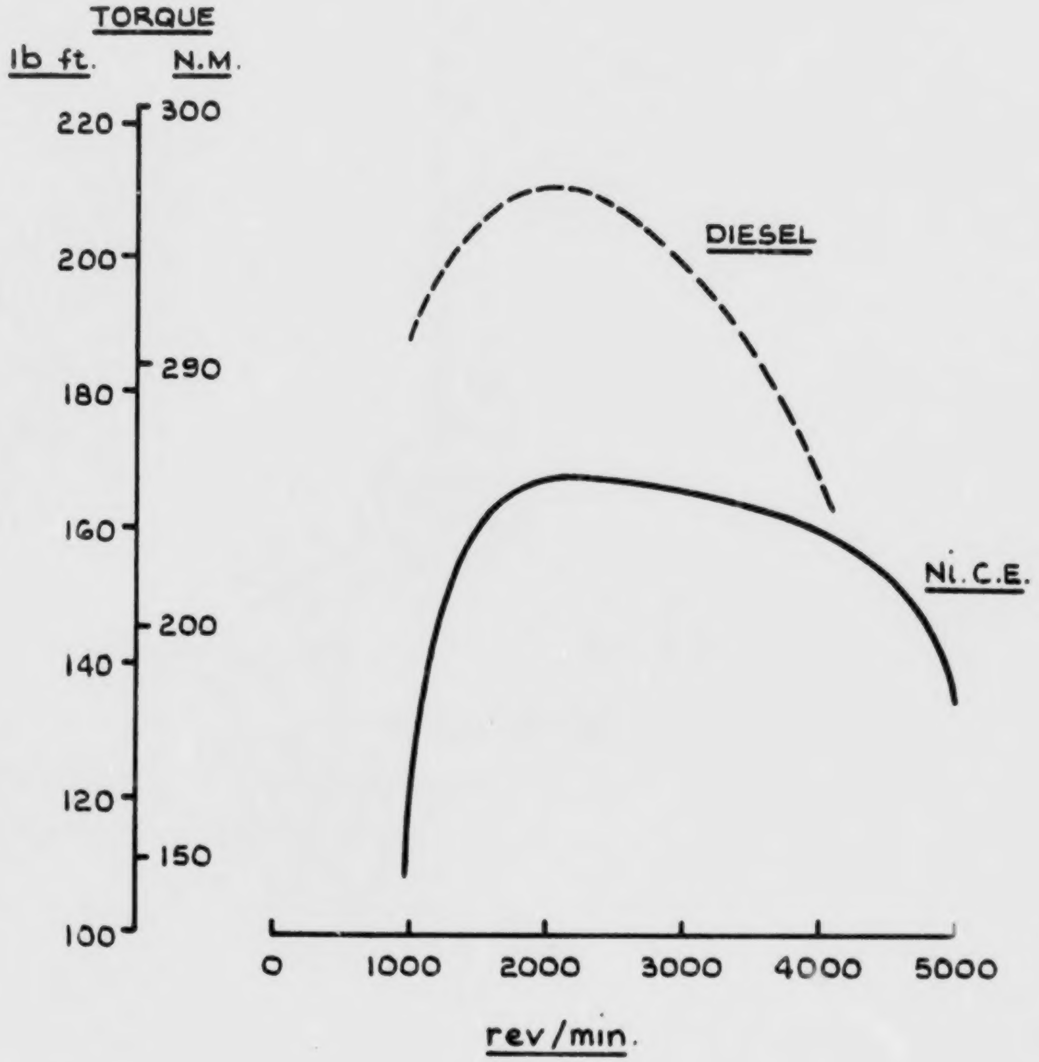


Figure 7.18. Comparison of torque curves for Kushul engine and baseline diesel engine.

KUSHUL ENGINE

SWEPT VOL. FOR 96 kW = 4·13 LITRES (254 in^3)
MAX. POWER SPEED 4000 rev/min.

DIESEL ENGINE

SWEPT VOL. FOR 96 kW = 4·77 LITRES (291 in^3)
MAX. POWER SPEED 4000 rev/min.

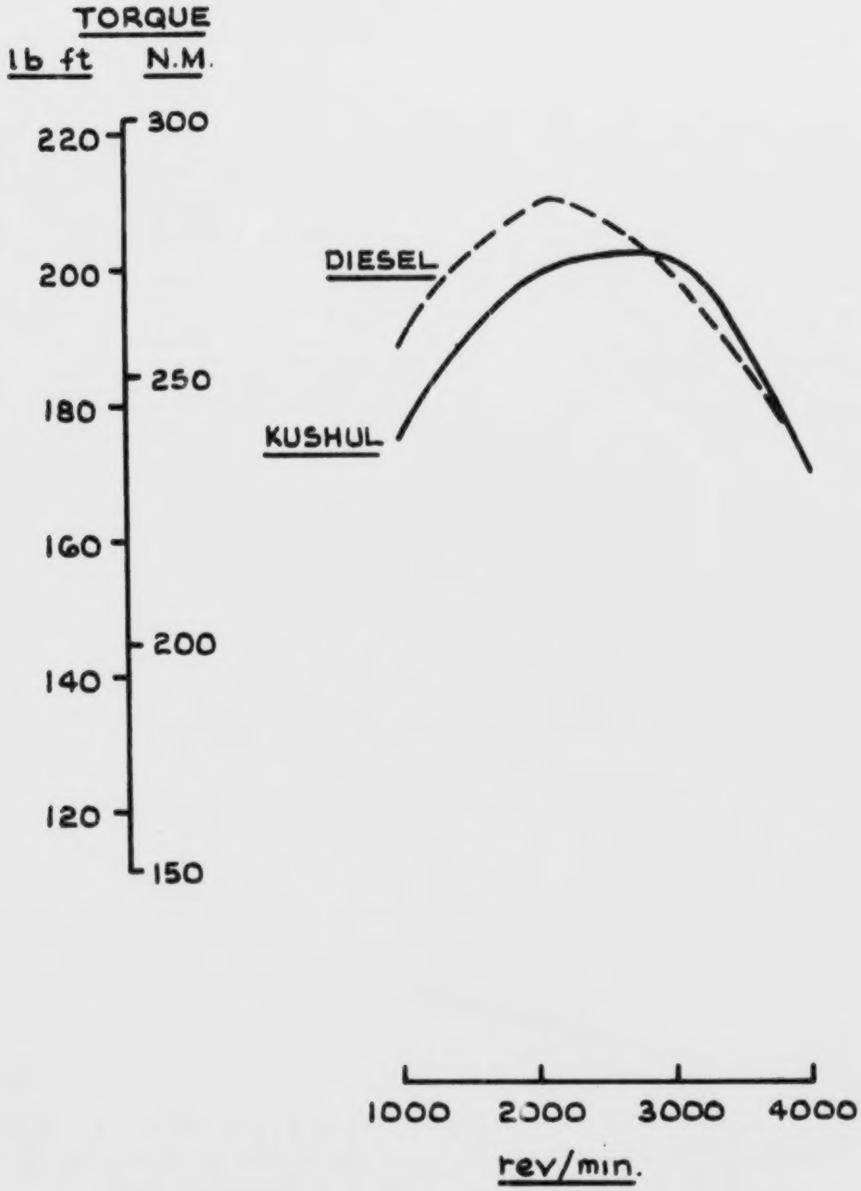
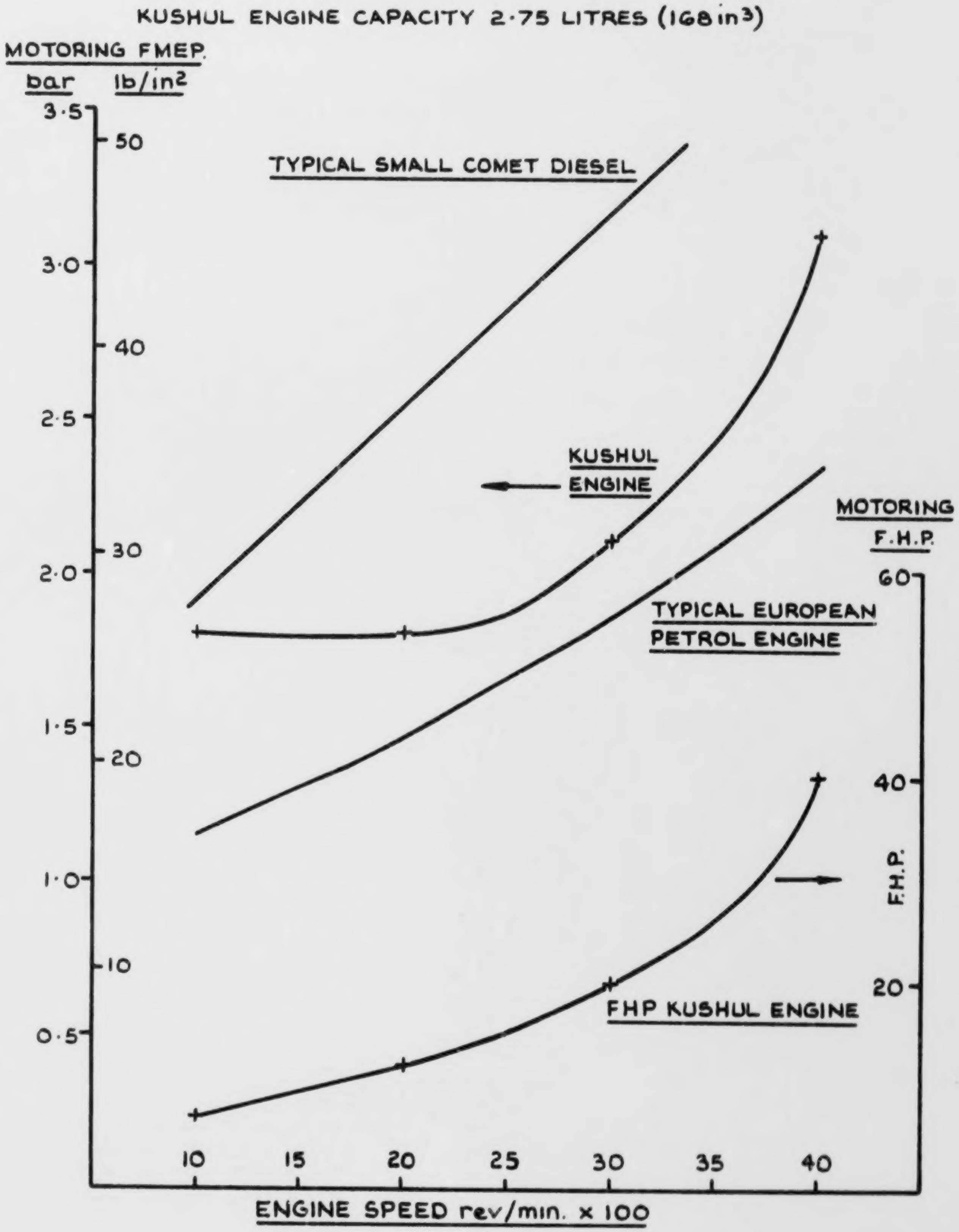


Figure 7.19. Motoring friction of Kushul engine in comparison to diesel and petrol engines.



Discussion on Results of Literature Survey as it Influenced the Configuration Study

An integral part of this project involved a study of the literature survey to decide which combustion systems might be viable for application to engines. A requirement for these power plants was that they should meet the specifications of the target vehicle, both with respect to exhaust emissions and performance. All possible engine configurations were to be considered, including turbocharged and rotary versions. In order to conduct this configuration study the examples in each category of stratified charge were inspected.

Category I. The Ford PROCO combustion system was the most developed example in this category, and had already been applied to a vehicle. Therefore this was chosen for the configuration study. The unthrottled version, FCP, would have great difficulties in meeting the HC emission target, even with catalysts. The other combustion systems in this category had not reached a sufficient stage of development to warrant an engine design.

The PROCO system was applied to both the primary and secondary emission targets.

Category II. The TCCS concept has also been applied to vehicles, and met the secondary emission targets, but the literature survey did not uncover any examples at the primary target. A conclusion from this survey was that the exhaust temperatures were too low for high catalyst efficiencies, due to the high operating air/fuel ratios of this concept. The only method by which the TCCS concept could comfortably meet the primary target would be by using larger quantities of EGR at low loads, to slow down the combustion and increase the exhaust temperature. Therefore, to bring the HC within 0.41 g/mile, the NO_x level would have to be reduced below 1.5 g/mile, probably to around 0.8 - 1 g/mile. TCCS engines were configured on this basis.

The TCCS concept was also adopted as a possible contender at the secondary emission target, using larger quantities of EGR to reduce the NO emissions.

The Curtiss-Wright was the only example of a rotary stratified charge engine uncovered by this survey, and it was considered desirable to produce a design scheme for this engine, in order that power plant advantages with respect to weight and size might be identified. However, this combustion system had not previously been applied to a vehicle, and it was appreciated that the HC and CO emission targets might be difficult to achieve.

There was no information relating to the use of EGR with this concept, so the secondary emission targets were not considered.

Category III. The MAN-FM combustion principle has been applied to a number of military and truck vehicles, but never to a car. The principle was adopted for an engine design, although a number of problem areas were known to exist (see Conclusions Category III) and these could not be solved at the design stage, but required engine development. Again there was no information relating to low NO levels, so the secondary targets were not considered.

Category IV. A large number of combustion systems were uncovered in this category but the results from the VW and Porsche systems were the most encouraging, particularly with respect to the inherent low NO emissions. Unfortunately, the Porsche arrangement could not be adopted, as this development is recent and insufficient information has been released to design an engine. Therefore, the VW concept was chosen for the primary targets. As the baseline NO emissions from this combustion system are low, it was considered that the same engine could probably meet the secondary emission target by the use of a suitable modulating EGR system, and extra catalyst control.

Category V. The Honda CVCC principle was adopted as typical of this category, and applied to meet the primary targets. Experimental evidence suggests that the secondary targets could also be achieved, using a larger engine.

Category VI. The advantages of the IFP process were considered to be rather small compared to the gasoline engine, particularly in view of the extra engine complexity. Therefore, this process was not adopted for a configuration study.

Category VII. In this category only the Kushul engine results were sufficiently attractive to justify further study. However, the two studies of this concept in England and Russia appeared to come to radically different conclusions about the specific power output. Without definitive figures on the performance of this engine it was difficult to conduct a design study. In addition, the exhaust emission potential of the engine had not been identified in detail. It was therefore decided not to pursue a configuration study.

Conclusions from Literature Survey

The following combustion systems were chosen for the configuration and rating study:-

- Ford PROCO
- Texaco TCCS
- CURTISS-WRIGHT ROTARY
- MAN-FM
- VW
- Honda CVCC

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